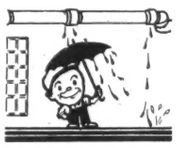
The EDITOR'S PAGE

Heating · Piping and Air Conditioning

THIS MONTH—

The first article in this issue is F. E. Wertheim's "Leak-Tight Joints—600 to 550,000 Lb." In it the author describes a number of types of pipe joints for use with high-pressure steam, air and other fluids which have demonstrated their reliability under operating conditions. Design of joints which will remain tight in service is a



major problem and the aim of Mr. Wertheim's article is to present helpful data concerning them. Following this informative paper, a brief article on page 473 tells how a temperature control problem involving an instantaneous heater supplying process water—steam being supplied from a turbine—was satisfactorily solved.

The next article, written by H. C. Weller, is "Utilizing Waste Gases for Factory Heating." The Gem Clay Forming Company saves $7500.00 per year by using waste heat from a continuous pottery kiln for heating and drying. The system is described by Mr. Weller, and a discussion on comfort and health in a plant heated in this manner is included, as is material on the necessary computations for determining the heat available.

Misconceptions as to what the Hartford loop is and what it does are prevalent; Robert E. Hattis endeavors to clarify the function of this piping arrangement in his interesting paper which starts on page 483. He shows an adapted form of the loop which he has used on a number of jobs. Mr. Hattis' article is followed by George W. Ott's description of the ventilating features of the new Chicago post office, an outstanding mail-handling center. This is an "air-rights" building and removal of locomotive smoke from the railroad tracks under the building is one of the ventilation problems.

Junius D. Edward's article on page 478 tells of the heating, piping and ventilating problems met with in a modern industrial research laboratory, and describes the Aluminum Research Laboratories; such factors as removal of fumes, flexibility

of piping, etc., are of prime importance in such buildings. Following this laboratory article, Malcolm Tomlinson shows how drying economy is aided by vapor proofing drying rooms.

In "On the Job" several short articles of particular interest to the operating, installing and maintenance engineer are published. The importance of operating valves with care is discussed by W. H. Wilson; need for the proper design of refrigerating coils is Robert S. Wheaton's subject; George T. Condron points out possible economies that may be obtained by the installation of a vacuum pump; John A. Masek describes an installation for conditioning paper.

There's no depression as far as the developments in air conditioning, heating and piping equipment are concerned; starting on page 526 of this issue a number of interesting developments and improvements are described. There are several new room coolers; improved valves; some pumps you'll want to know about and a number of other devices.

NEXT MONTH—

Of interest to engineers concerned with heating is the paper which has been prepared by C. E. Joos and V. A. Rohlin on corrosion of hot water lines and its correction; divided into several installments, the first of this valuable series will appear next month. Use of deareating equipment is discussed in detail.

Oil-handling stations and many industrial plants require provisions for heating heavy oils so that they may be handled quickly and economically. In his article to appear in the August issue, Samuel R.

Lewis discusses piping steam for heating oil tanks, describing effective piping lay-outs and pointing out the factors to consider.

In high-humidity rooms in paper mills, weaving rooms, dye houses, etc., a properly insulated roof to eliminate condensation and drip in order to prevent spoilage is of importance. In an article by Ralph Winslow to be published next month, a chart for selecting the proper thickness of insulation is presented and the experience of two mills is described.

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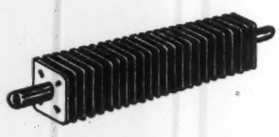

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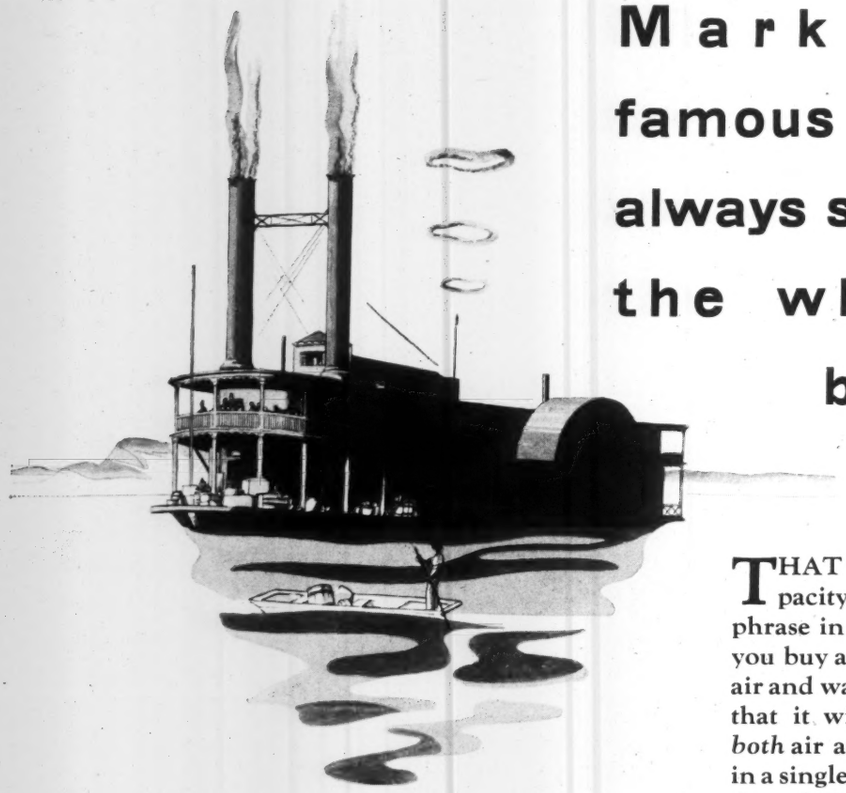

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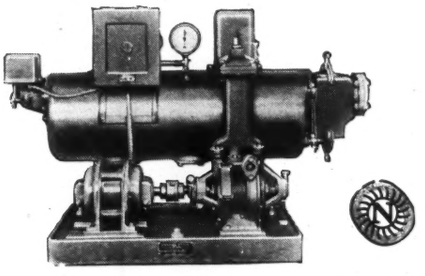

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Heating · Piping and Air Conditioning

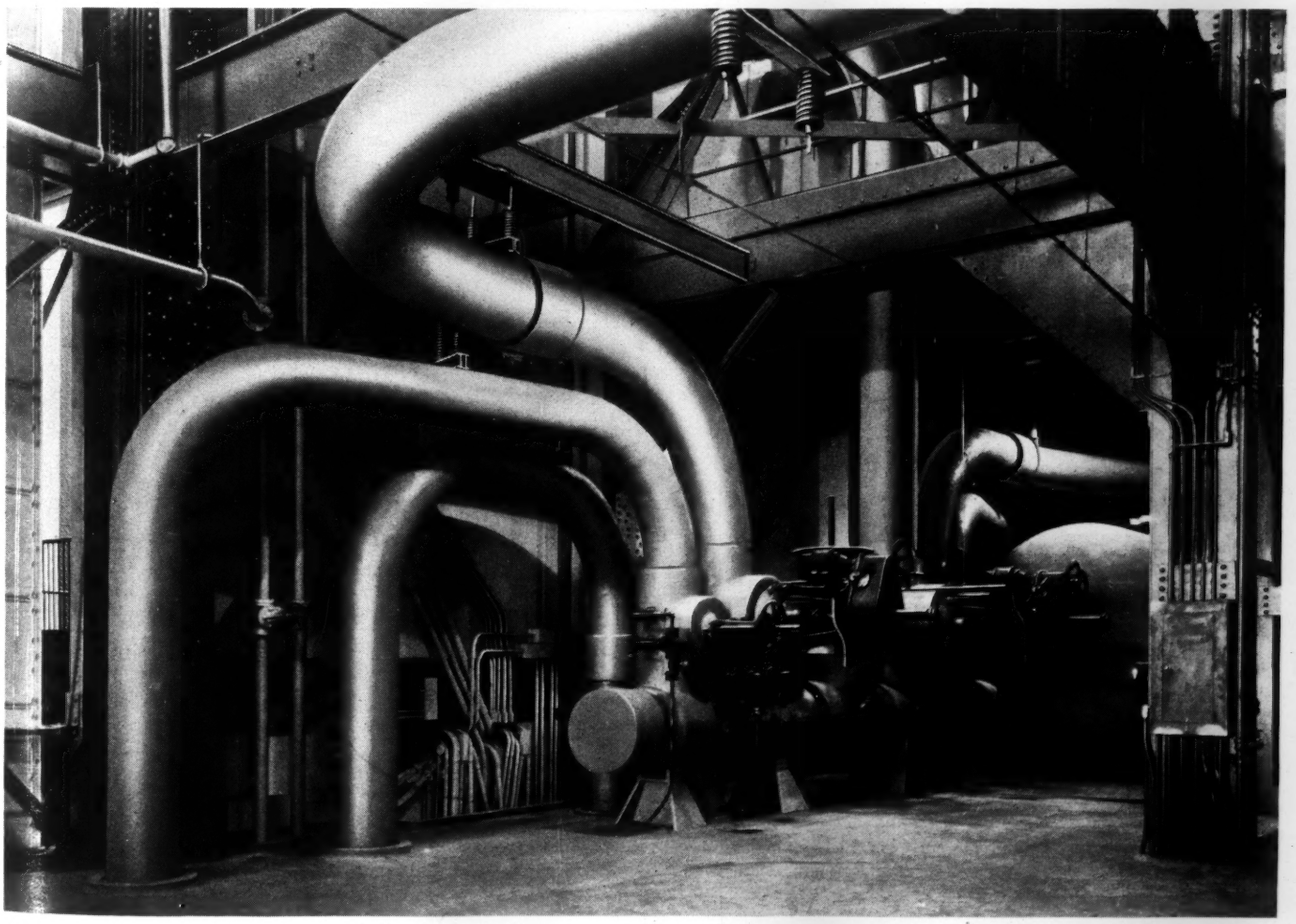
Vol. IV July, 1932 No. 7

Leak-Tight Joints — 600 to 550,000 Lb.

By F. E. Wertheim*

TIGHT joints which will remain tight are the only ones which are considered satisfactory and the design of such joints is a major problem. It is the purpose of this article to discuss various types of high-pressure joints which have successfully met the conditions imposed by different classes of service and to present data concerning them; a comprehensive outline touching upon the advantages offered by different types of construction and presenting some recent developments

*Associate Member, *A. S. M. E.*



MAIN STEAM LEADS AT MANIFOLD, GREAT WESTERN POWER CO., SAN FRANCISCO, CALIF. CONSULTING ENGINEERS, MCCLELLAN & JUNKERSFELD, INC.

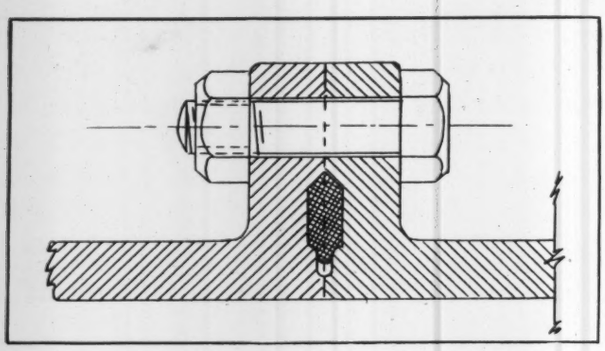


FIG. 1—AN AIR-TIGHT JOINT WHICH HAS BEEN USED FOR PRESSURES OF 2000 LB. PER SQ. IN.

is the aim. Information pertaining to the requirements of high-pressure joint design as regards strength, tightness, and creep is available by reference to articles which have appeared in these columns; these are listed in the bibliography, and repetition has been avoided.

Air-Tight Joints for 2000 Lb. Per Sq. In.

Necessity for high air pressures began with the development of apparatus for producing liquid air, and with the use of high-pressure air for operation of early types of Diesel engines. Another early application of high-pressure air was made by the U. S. Ordnance Department, which had compressors built with pressure capacity of 2,000 lb. per sq. in. as propulsive force for the Zalinski[1] pneumatic dynamite gun.

For the gun sections, for piping and various other connections, the joint shown in Fig. 1 was invented by Captain Rapieff, U. S. A. This is a metal-to-metal joint with a groove turned in each flange, overflow space being provided beneath the soft rubber packing, which consists of a length of round rubber with scarfed ends. The gasket is entirely enclosed, and escaping air drives the rubber into the angle at the top, thus sealing off leakage. This construction was used in a large number of cases and proved highly satisfactory (*12*).[2]

Water-Tight Joints Under 1750-Ft. Hydraulic Head

Fig. 2 shows a joint which has been used for hammer-welded penstock pipes, the one for the San Gorgonio Power Company, at Banning, Cal., operating under 760 lb. per sq. in. pressure. It will be noticed that this is a square corner lap metal-to-metal joint, with thickness of ends heavier than the pipe wall, an entirely confined gasket, and a minimum diameter of bolt circle. Due to this construction the advantages of a minimum of friction, a tight joint, minimum flange bending stresses, and economy of material are secured. Joints of this type are operated under pressures as high as 2,170 lb. per sq. in.

As noted in a recent number of this magazine (*13*) joints with confined red fiber gaskets are quite applicable to general use in connection with high water pressure.

[1] A battery of these was at one time mounted in Ft. Winfield Scott, San Francisco.

[2] See Bibliography in next month's issue.

Gas-Tight Joints at 800 Lb. Per Sq. In.

Dresser-type couplings, (Figs. 3, 4 and 5) were originally designed for high-pressure natural gas lines, but their use has been extended to include penstocks and flumes, also water, oil, compressed air, vacuum, brine, sewage, and manufactured-gas lines. This type of joint may be used on any kind of plain-ended pipe in sizes up to 5 ft. 6 in. diameter and larger. We are here concerned mainly with its suitability for high-pressure gas-line service.

In long-distance transportation of natural gas, high-pressure lines consisting of long lengths of pipe made up by welding shorter joints are laid, the long sections (about 80 ft.) being united with such couplings. The long sections increase the holding power of the ditch, protecting the line against washouts, while the couplings permit flexibility, prevent leakage, and provide for expansion and contraction due to temperature changes. There is a rolling action due to resiliency of the packing, rather than slippage between rubber and pipe.

With electric weld and seamless tubing available, pressures have been increased to stress the line to the maximum considered safe, 800 lb. per sq. in. being about the present limit.

Tests have been made on couplings of this type (*14*) to determine the bolt stresses required to hold various line pressures and the amount of end movement which the rubber could take care of without slippage. (Slippage would result in injury to packing and leakage.) In the tests the pipe was deflected 4 deg. while under 1,600 lb. per sq. in. pressure, without leakage. A 24-in. pipe coupling subjected to gas pressure while submerged in water withstood 2,250 lb. per sq. in., and leakage did not occur when pipe ends were rough and pitted.

Design, based on data secured from the foregoing tests, has resulted in couplings that will satisfy the requirements of extremely severe field conditions. For heat conditions, special gaskets of plain rubber have been furnished for temperatures up

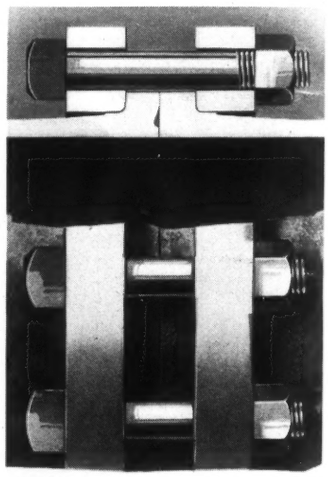


FIG. 2—A PENSTOCK JOINT USED ON HYDRAULIC HEADS OF 1,753 FT. (760 LB. PER SQ. IN.). NOTE THE RESEMBLANCE TO STEAM-LINE JOINTS

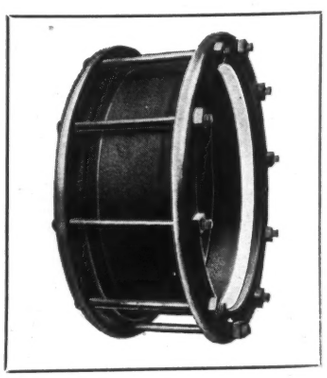


FIG. 3—DRESSER-TYPE COUPLING

Design of joints for high-pressure piping which are tight and will remain tight is a major problem. In this article several types of joints which have successfully met the conditions imposed by various kinds of service are described and recent trends in the design of such joints are discussed.

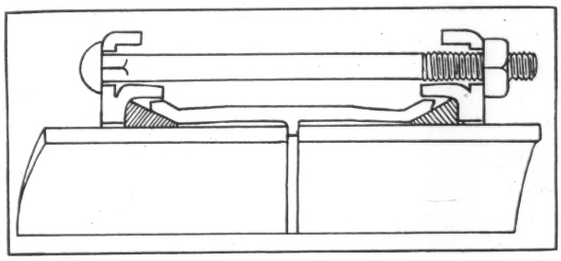


FIG. 4—THE JOINT SHOWN IN FIG. 3 ASSEMBLED

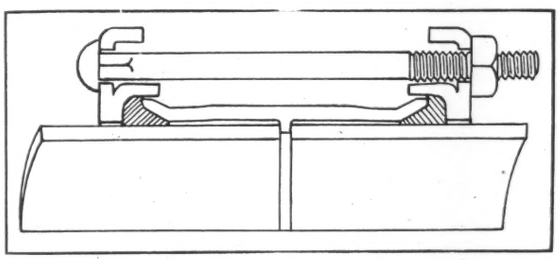


FIG. 5—SHOWING THE GASKET COMPRESSED (SEE FIGS. 3 AND 4)

to 350 F; it is claimed that special asbestos gaskets will function for temperatures up to 1,000 F.

Split-Housing Flexible Joints

Another type, widely used for the same purposes as the foregoing, is the split-housing flexible joint. Fig. 6 shows a grooved pipe end, the more usual construction, though several kinds of shouldered ends are also in use. This joint provides for angular deflection, expansion, and contraction of pipe lengths, and locking the pipe ends together, the latter feature obviating the need of anchorages. The depth of grooves is just a trifle less than that of standard pipe threads.

For pressures of 800 lb. per sq. in. or higher, forged housings are in general use, those for 14-in. pipe and larger being in four or more sections. A special resilient sealing ring spans the pipe ends and is confined by the housing which is bolted over it. As shown, the sealing ring lip extends downward, and when the ring gasket is put on the pipe the lip is straightened out, providing the necessary seal. Increase of line pressure forces the lips more tightly against the pipe, increasing the tightness of the joint.

In a test (*15*) a 10-in. pipe equipped with split-housing couplings remained tight under a hydrostatic pressure of 1,700 lb. per sq. in.

Steam Tight Joints—600 Lb. W.S.P. to 3300 Lb. W.S.P.

In considering steam-tight joints it will be interesting to examine two foreign designs, the Loeffler and Benson, before proceeding with discussion of American types.

The Loeffler joint, first applied to flanged pipe connections, was thereafter used by Professor Loeffler for all connections to his high-pressure boiler and its associated apparatuses. Referring to Fig. 7, it will be noted that this joint is of plug-and-socket construction, the plug provided with a narrow shoulder acting as a depth stop. Both plug and socket are slightly tapered, the diameter of the former being such that bolting up the joint causes sufficient deformation of the sealing surfaces to form an impervious connection. Because of the taper force-fit, these joints may be repeatedly disconnected without injury to the sealing surfaces. Where exposed to sudden temperature changes, as in non-return valves, the sealing cones are protected by insertion of sheet-metal funnel sleeves which prevent access of steam to joint surfaces.

It was found that without this sleeve protection any rapid fluctuations of steam, especially with the high superheat used, caused warping and leakage. Use of studs, instead of bolts, is intended to facilitate heat trans-

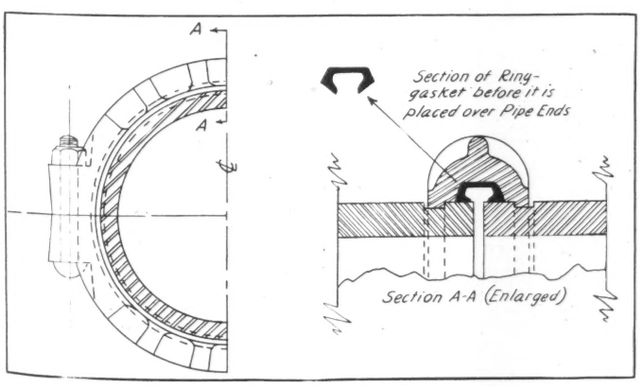


FIG. 6—SPLIT-HOUSING FLEXIBLE JOINT FOR AIR, GAS OR WATER (PRESSURE, 800 LB. PER SQ. IN.)

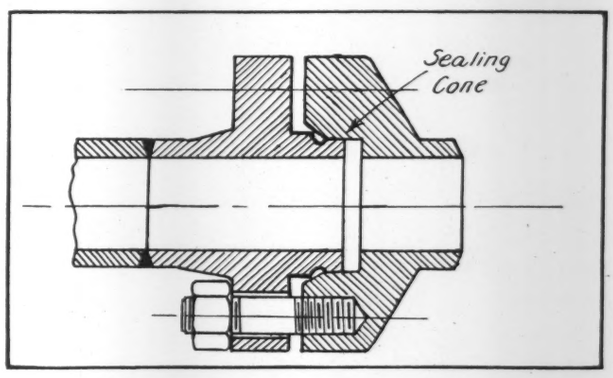


FIG. 7—LOEFFLER STEAM-TIGHT JOINT WHICH HAS BEEN USED ON PRESSURES OF 1,900 LB. PER SQ. IN. AND 930 F

fer and to reduce differential expansion between studs and flanges.

This joint has been used in sizes up to 10 in. I. D. for pipe, and over 12-in. diameter for cover plates, and it is stated that still larger diameters are practicable. In making hydrostatic boiler tests no leakages occurred with pressures ranging from 3,750 lb. to 4,500 lb. per sq. in. Continued dependability has been demonstrated by several years of service at 1,900 lb. W. S. P. and 930 F.

The Benson joint shown in Fig. 8 was first applied to carbon steel boiler tubes, 0.78 in. I. D. and 1.26 in. O. D., with pressure of 3,380 lb. per sq. in. at 842 F. The same construction was later employed in a larger size for the steam piping of the turbine, operated at 1,470 lb. W. S. P. and 750 F.

The hubbed flanges are loosely screwed on the pipes, and a packing (shown shaded) bearing directly on the pipe ends seals the joint. Long bolts were purposely used with springs placed beneath the nuts to provide for the large temperature variations, these springs retaining their full elasticity with 840-F steam in the pipe. It is claimed that these joints were tight and satisfactory.

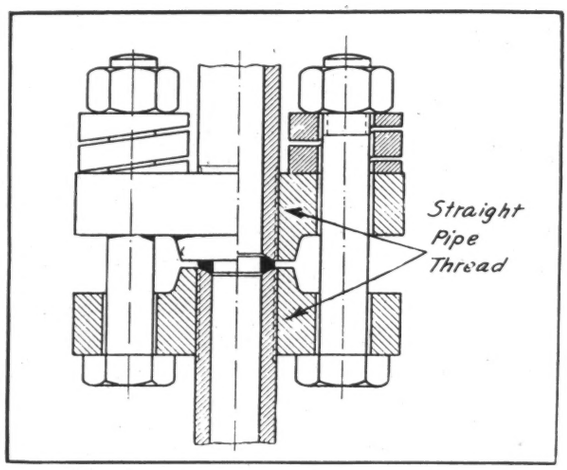


Fig. 8—Joint for Benson Boiler Used on Pressures of 3,380 Lb. Per Sq. In. and 840 F

Weld-Seal Joints

In American practice many steam plants, and especially those operating under high pressures and temperatures, have employed weld-seal joints. Attention is directed to Fig. 9 which shows (with some exaggeration in the drawing) recommendations of Sargent and Lundy, Inc., consulting engineers, for machining weld-seal surfaces. The

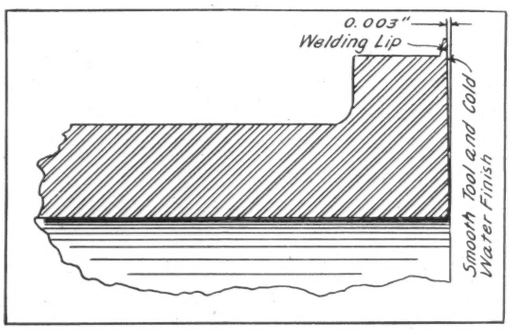


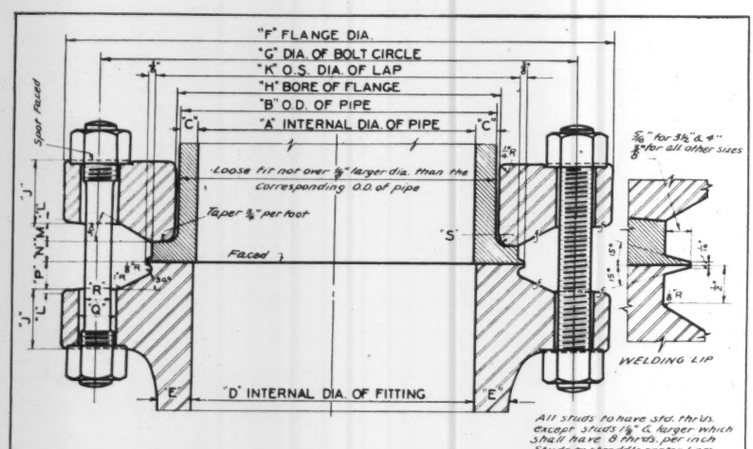
Fig. 9—Weld-Seal Joint for 1500 Lb. Per Sq. In. Note Recommendations for Machining the Contacting Faces

contact faces should be very slightly dished, so that the opening at the outer edges of the lips is about 0.006 in. when the pipe laps are brought to contact at the bore. Because of the decrease in contact areas thus obtained, some deformation undoubtedly takes place when the studs are pulled up, and slight deformation of surfaces or of gaskets seems to be essential for tightness in high-pressure joints.

It is now generally known that the 1,350-lb. pressure rating is to be changed to 1,500 lb. W. S. P. at 750 F, although the dimensional tables have not yet become available. Fig. 9*A* and its accompanying table give the Sargent and Lundy standards for 1,500 lb. W. S. P., for sizes up to 24 in., as applied to weld-seal joints.

Dimensions of weld-seal fittings differ from those of the *A. S. A.*, first because of the special facing, and second because the addition due to "Height of Lip on Fitting" results in a greater "Center to Flange Edge" dimension in the weld-seal fittings than in the *A. S. A.*

In some hydrostatic tests made on gasket-


"F" FLANGE DIA.
"G" DIA. OF BOLT CIRCLE
"K" O.S. DIA. OF LAP
"H" BORE OF FLANGE
"B" O.D. OF PIPE
"A" INTERNAL DIA. OF PIPE
Spot Faced
Loose fit not over 1/8" larger dia. than the corresponding O.D. of pipe
Taper 3/4" per foot
Faced
WELDING LIP
"D" INTERNAL DIA. OF FITTING
All studs to have std. thr'ds. except studs 1 1/8" & larger which shall have 8 thr'ds. per inch. Studs to straddle center lines.

NOMINAL SIZE	INSIDE DIA. OF PIPE A	OUTSIDE DIA OF PIPE B	THICKNESS OF PIPE C	INSIDE DIA OF FITTING D	THICKNESS OF FITTING E	FLANGE DIAMETER F	DIA OF BOLT CIRCLE G	MAX BORE COMP FLANGE H	THICKNESS OF FLANGE J	OUTSIDE DIA OF LAP K	L	M	N	P	Q	R	S	NO. OF STUDS	DIA OF STUDS	CENT TO FACE OF TEES & CROSSES	CENT TO FACE 45° ELL	LONG CENTER TO FACE OF REDUCERS	SHORT CENTER TO FACE REDUCERS
2½	2⅛	2⅞	⅜	2¼		9⅝	7½	3	1⅞	4⅛	1/8	5/16	⅜	11/16	1/8	13/16	⅜	8	1	13	10	17¼	10
3	2⅝	3½	7/16	2¾		10½	8	3⅜	2⅛	5	⅛	5/16	7/16	13/16	⅛	1¼	13/16	⅜	8	1⅛	14¾	11¼	19¼	11¼
3½	3⅛	4	7/16	3¼		11	8½	4⅛	2⅜	5½	⅛	5/16	7/16	13/16	⅛	1⅜	13/16	⅜	8	1⅛	16	12	21	12
4	3½	4½	½	3¾		12½	9½	4⅝	2⅝	6⅛	⅛	⅜	½	¾	⅛	1⅜	1 1/16	½	8	1¼	17½	13½	23	13½
5	4⅛	5⅛	½	4⅝		14½	11½	5⅛	2⅞	7⅛	⅛	⅜	½	¾	⅛	1⅜	1 1/16	½	8	1¼	20	14½	26½	14½
6	5⅛	6⅝	½	5⅝		15½	12½	6⅝	3⅛	8½	⅛	⅜	½	1	⅛	1½	1 1/16	½	12	1⅛	21½	16½	29½	16½
8	6⅞	8⅝	⅝	7		19	15½	8¾	3⅜	10⅛	⅛	⅜	½	1⅛	⅛	1¾	1 1/16	½	12	1⅜	25½	18½	34½	18½
10	8⅞	10¾	1 1/16	8¾		23	19	10⅞	4⅛	12¾	⅛	⅜	½	1⅛	⅛	2	1 1/16	½	12	1⅝	29½	21½	42	21½
12	10⅞	12¾	1 1/32	10⅝		26½	22½	12⅝	4⅝	15	⅛	⅜	½	1 1/32	⅛	2⅛	1 1/16	½	16	2	33	24½	49	24½
14	11⅞	14	1 1/32	11⅝		29½	25	14⅛	5⅛	16½	⅛	⅜	½	1 1/32	⅛	2⅜	2 1/16	½	16	2¼	36¼	25½	51¼	25½
16	13	16	1 1/32	13		32½	27½	16⅛	5⅝	18½	⅛	⅜	½	1 1/32	⅛	2⅝	2 1/16	½	16	2½	40	28½	59	28½
18	14⅛	18	1 1/16	14⅝		36	30½	18⅛	6⅛	21	⅛	1	1 1/16	1 1/32	⅛	2⅞	2⅝	½	16	2¾	44½	31½	66	31½
20	16⅛	20	1 3/32	16⅝		38¾	32¾	20⅛	7	23	⅛	1⅛	1 3/32	1 5/16	⅛	3⅛	2 1/16	½	16	3	48½	34¾	71½	34¾
24	19⅛	24	2 1/16	19⅛		46	39	24⅛	8	27¼	⅛	1⅛	2 1/16	1 5/16	⅛	3⅜	3⅝	½	16	3½	57	40	82¾	40

METAL THICKNESS TO BE GREATER THAN AMERICAN STD., CONTRACTOR TO SUBMIT THICKNESS OF FITTING WITH PROPOSAL

Fig. 9*A*—Standards for 1500 W.S.P. Weld-Seal Joints

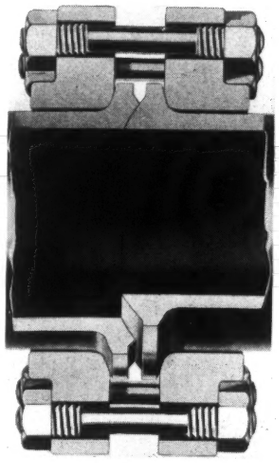


FIG. 10 — SPHERICAL-SEAT JOINT, PIPE-TO-PIPE APPLICATION

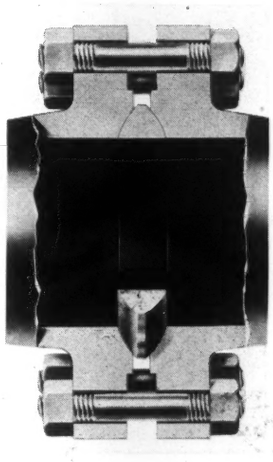


FIG. 11 — SPHERICAL-SEAT JOINT, FITTING-TO-FITTING APPLICATION

less weld-seal joints, a 12-in. series 150 joint before welding failed at a pressure of 16,000 lb. per sq. in., remaining tight up to 10.67 times the working pressure and the joint after welding failed at a pressure of 22,000 lb. per sq. in., or 14.67 times the working pressure. Bolt stresses when leakage occurred were 31,400 and 43,200 lb. per sq. in. respectively.

The method of erecting piping fitted with joints of this construction is to assemble the piping, pull up the stud bolts, make hydrostatic test (followed by steam test when so specified), and when tightness has been proved to remove one bolt at a time and weld the lips.

Spherical-Seat Joints

Another type of joint in which the gasket is eliminated is the one shown in Fig. 10, in which all lap and flange contact surfaces are machined to spherical shape. The joint contact for pipe ends is between concave and convex surfaces, while valves and fittings are made concave as in Fig. 11, where a loose steel ring closes the joint. These joints are now in use for pressures of 900 lb. per sq. in. and fluid temperature of 1,100 F.

This type is particularly applicable to inaccessible locations where it is difficult to replace gaskets. It remains tight when the piping is not perfectly aligned, the permissible variation of about 3 deg. simplifying installation.

Note: In the second part of this article to appear in next month's HEATING, PIPING AND AIR CONDITIONING, Mr. Wertheim discusses *A. S. A.* tongue-and-groove joints, ring joints, wedge ring joints, a union for small size high-pressure tubing used by the Fixed Nitrogen Research Laboratory of the Department of Agriculture, and a joint for large-diameter apparatus. An important part of next month's installment is devoted to leak-tight joints at 550,000 lb. per sq. in., involving Dr. Percy W. Bridgman's unsupported area principle as applied to research work at this tremendous pressure.

Solving a Temperature Control Problem

By E. D. Rogers*

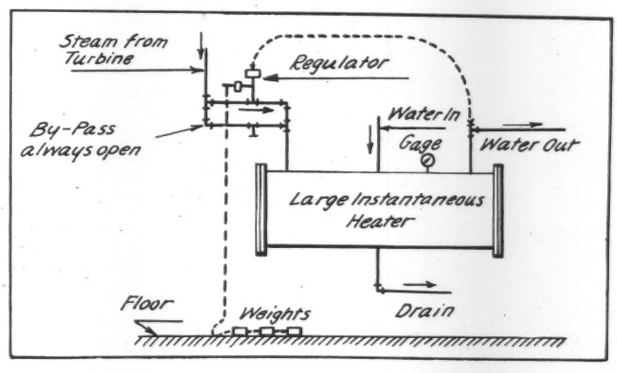
IN A large manufacturing plant, a 6-in. temperature regulator was installed on an instantaneous heater which drew its steam supply from a turbine. When the regulator would close, a 26-in. vacuum would be generated in the heater and when the regulator would open it opened wide, thus throwing a heavy load on the turbine. To overcome this, the plant engineer rigged up the apparatus shown in the accompanying drawing.

A chain was attached to the regulator lever and three weights were fastened to the chain so that normally the weights rested on the floor. (Position of weights and slack in chain is slightly exaggerated on drawing for clearness sake.) When the lever would raise suddenly as the valve opened, the slack chain picked up these weights one at a time, thus retarding the opening of the valve and allowing steam to enter the heater gradually. This operation was, of course, reversed on the closing stroke of the valve.

*Fulton Sylphon Co., Knoxville, Tenn.

By-Pass Kills Vacuum in Heater

This method worked very well but later a 1¼-in. by-pass around the regulator was substituted for the system of chain and weights. This by-pass is kept open at all times and kills the vacuum in the heater and softens the operation of the regulator which now seldom closes off entirely. In this particular plant, the heater is in operation 24 hours every day and large quantities of water are used; consequently, the by-pass never causes any overheating.



METHOD OF CONTROLLING A HEATER SUPPLYING PROCESS WATER. STEAM SUPPLY IS DRAWN FROM A TURBINE

By H. C. Weller*

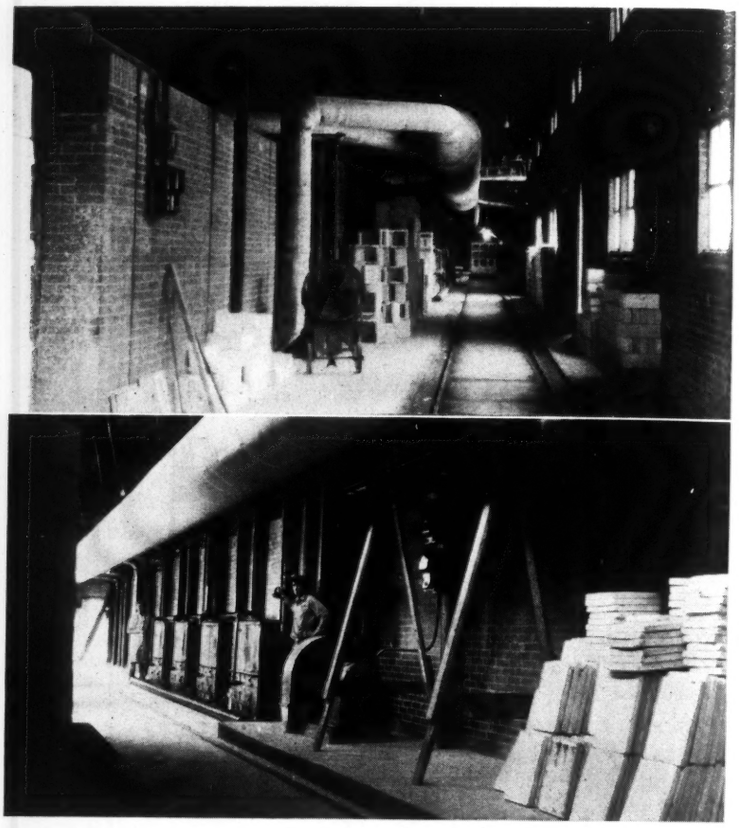


FIG. 1—TWO VIEWS OF THE MAIN TRUNK LINE ALONGSIDE THE KILN, WHICH ACTS AS A STORAGE VESSEL FOR THE WASTE GASES

UTILIZING Waste Gases for Factory Heating

A TOTAL saving of $7,500.00 per year in the cost of heating its building is made by the Gem Clay Forming Company of Sebring, Ohio, by utilizing waste heat from its continuous pottery kiln for factory heating and drying. The savings amount to $25.00 per day during the heating season and $18.00 per day when the dryers only are in operation. This company is engaged principally in the manufacture of radiant fire backwalls, radiants and saggers for ceramic uses. In addition to these, ornamental clay animals, vases and pins for potteries are made on a smaller production scale. The capacity of the plant is about three thousand backwalls and at least forty thousand radiants per day.

Continuous Kiln Fired with Natural Gas

These products were at first fired in up-draft upright kilns of conventional design. However, it was later found necessary to build a continuous car type tunnel kiln in order to keep pace with increasing demand for the product. The time of firing backwalls and radiants varies between forty and fifty hours, chiefly depending upon grade and size of ware as well as various other factors. The continuous kiln is 275 ft. long and fired with natural gas. No muffle is necessary in the firing chamber since combustion is automatically controlled and thus the temperature in the firing zone is constantly held at 2200 F.

Average Gas Consumption 6,000 c.f.h.

The average consumption per hour of this kiln on the normal operating schedule is 6,000 cu. ft. of natural gas, chiefly depending upon car speed, amount, size and ingredients of the ware passing through the kiln. The amount of induced air at the burners necessary for complete combustion of the natural gas is 10.2 times the quantity of gas. In firing this class of ware, a strictly oxidizing kiln atmosphere is desirable. During the mixing of the various clays and other ingredients, a definite portion of sawdust is added which burns out while the ware is fired and renders the backwalls and radiants porous. A certain humidity in the preheating zone of the kiln is always maintained and controlled because of the danger of surface cracking of the highly intricate patterns which were previously colored before entering the kiln.

The plant was at the time of completion of the continuous kiln heated by two 35-hp. boilers which were in

*Industrial engineer, Salem, Ohio.

severe weather insufficient chiefly because of proportionally increased heat demand for drying purposes. Naturally, the question arose whether or not the waste heat from the continuous kiln could be made available to serve both purposes, *i.e.*, heating of plant and office buildings and drying of ware.

By utilizing waste heat from a continuous pottery kiln for factory heating and for drying, the Gem Clay Forming Company saves $7,500.00 annually.

The Fundamental Principles Involved

Fundamental principles involved in the consideration were the determination of the amount of heat energy which could be utilized from the waste gases, the effect of these gases, their humidity and movement in regard to comfort and health conditions in the plant and, of course, the mechanical principles of the heating plant itself.

Determination of the Available Heat in the Flue Gases

If the components of the fuel gas are known, the volume of the products of combustion and the heat available in these products can be calculated. Assume a natural gas, having the following analysis (see *A.G.A.* Handbook, 1926);

	Per cent by volume	Fraction of a cu. ft. in a cu. ft.
Methane CH_4	84.6	0.846
Ethane C_2H_6	13.4	0.134
Nitrogen N_2	2.0	0.020 (inert)
	100.0	1.000

In order to determine the amount of air necessary for the combustion of one cu. ft. of this gas, it is necessary to determine the volume of air required for each of the combustible elements and then add up these volumes. The volumes of air necessary can be calculated directly from the combustion equations of the various components or else by means of figures taken from a table to be found in engineering handbooks:

Air required for	Cu. ft. at 60 F and 30″ barometer
Methane 0.846×9.57	8.10
Ethane 0.134×16.75	2.24
Total	10.34 cu. ft.

Thus for every cu. ft. of the above gas burned, 10.34 cu. ft. of air must be supplied.

The volumes of the products of combustion can also be calculated directly from the combustion equation of the components:

Methane CH_4 plus $2O_2 = CO_2$ plus $2H_2O$
Ethane C_2H_6 plus $7O_2 = 4CO_2$ plus $6H_2O$

Thus, ethane yields twice as much carbon dioxide by volume as methane does which requires two volumes of oxygen for combustion and yields one volume of carbon dioxide. Hence,

Carbon dioxide from	
Methane 0.846×1.0	0.846
Ethane 0.134×2.0	0.268
Total CO_2	1.114 cu. ft.
Nitrogen from	
Gas (analysis)	0.020
Air (supplied for combustion)	10.340
Total N_2	10.360 cu. ft.

If the products of combustion of one cu. ft. of the above gas are cooled so that the water vapor condenses and are measured at the same temperature and pressure at which the gas was measured, they will consist of 1.114 cu. ft. of carbon dioxide and 10.360 cu. ft. of nitrogen, a total of 11.474 cu. ft.

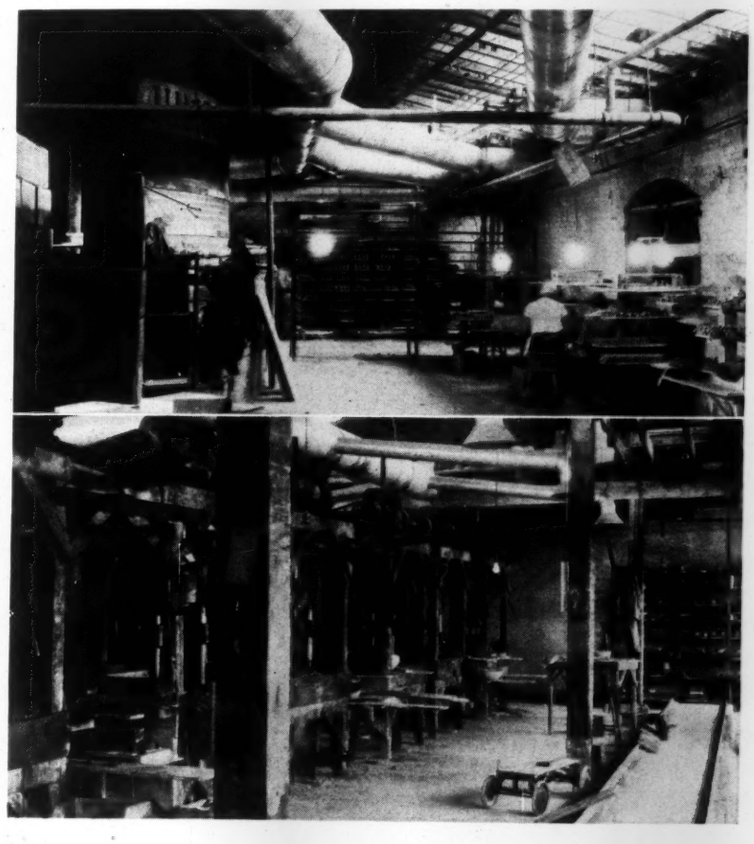
The extent to which the products of combustion are diluted by excess air determines, however, the temperature at which condensation of liquid water begins. In our application, the temperature of the flue gases was well above the critical temperature for condensation and there is, therefore, a volume of water vapor present which can be calculated from the above cited combustion equations:

Water vapor, H_2O, from combustion of	
Methane 0.846×2.0	1.692
Ethane 0.134×3.0	0.402
Total	2.094 cu. ft.

Total products of combustion per one cu. ft. of gas burned (no excess air)

Total dry products	11.474
Total water vapor	2.094
	13.568 cu. ft.

FIG. 2—A VIEW OF THE DISTRIBUTION LINES IS SHOWN ABOVE. BELOW—DISCHARGE LINES IN THE CLAY PRESS ROOM 600 FT. FROM THE CONTINUOUS KILN



The determination of the amount of excess air in the flue gases was carried out by means of a gas analysis taken with an Orsat apparatus and a continuous CO_2 recorder over a period of a few days. These samples were taken at a convenient point about 20 ft. distant from the discharge end of the kiln. It is quite apparent that a certain dilution of the hot gases coming from the firing zone had taken place. The mean analysis was as follows:

3.5% CO_2
9.4% O_2
87.1% N_2 and water vapor.
100.0%

The temperature of these flue gases was 440 F; the amount of excess air was calculated according to the following table:

TABLE FOR COMPUTING % EXCESS AIR IN FLUE GAS

According to formula:

$$\% \text{ excess air} = \frac{79.09 \times O_2}{(20.91 \times N_2) - (79.09 \times O_2)} \times 100$$

which reduces to:

$$\frac{O_2}{0.2644\ N_2 - O_2} \times 100$$

The product $0.2644\ N_2$ is given in the body of the table:

N_2	0.00	0.10	0.20	0.30	0.40	0.50	0.60	0.70	0.80	0.90
77.0	20.36	20.38	20.41	20.44	20.46	20.49	20.52	20.54	20.57	20.60
78.0	20.62	20.65	20.67	20.70	20.73	20.75	20.78	20.81	20.83	20.86
79.0	20.89	20.91	20.94	20.97	20.99	21.02	21.04	21.07	21.10	21.12
80.0	21.15	21.18	21.20	21.23	21.26	21.28	21.31	21.34	21.36	21.39
81.0	21.41	21.44	21.47	21.49	21.52	21.55	21.57	21.60	21.63	21.65
82.0	21.68	21.71	21.73	21.76	21.79	21.81	21.84	21.86	21.89	21.92
83.0	21.94	21.97	22.00	22.02	22.05	22.08	22.10	22.13	22.16	22.18
84.0	22.21	22.23	22.26	22.29	22.31	22.34	22.37	22.39	22.42	22.45
85.0	22.47	22.50	22.53	22.55	22.58	22.60	22.63	22.66	22.68	22.71
86.0	22.74	22.76	22.79	22.82	22.84	22.87	22.90	22.92	22.95	22.98
87.0	23.00	23.03	23.05	23.08	23.11	23.13	23.16	23.19	23.21	23.24
88.0	23.27	23.29	23.32	23.34	23.37	23.40	23.42	23.45	23.48	23.50
89.0	23.53	23.56	23.58	23.61	23.64	23.66	23.69	23.72	23.74	23.77
90.0	23.79	23.82	23.85	23.87	23.90	23.93	23.95	23.98	24.01	24.03
91.0	24.06	24.09	24.11	24.14	24.16	24.19	24.22	24.24	24.27	24.30
92.0	24.32	24.35	24.38	24.40	24.43	24.46	24.48	24.51	24.53	24.56
93.0	24.59	24.61	24.64	24.67	24.69	24.72	24.75	24.77	24.80	24.83
94.0	24.85	24.88	24.90	24.93	24.96	24.98	25.01	25.04	25.06	25.09

Example: To determine the % excess air in a flue gas having 9.4% O_2 and 87.1% N_2

First look in body of table for product of $0.2644N_2$ which is 23.03; subtract the O_2 from this value and then divide the O_2 by the remainder and multiply by 100.

$$\frac{9.4 \times 100}{23.03 - 9.4} = \frac{9.4 \times 100}{13.63} = 68.97\% \text{ excess air.}$$

The amount of excess air in the flue gases was calculated to be 70 per cent. Therefore, for each cubic foot of gas burned there is:

$$13.568 + (0.10342 \times 70) = 20.807 \text{ cu. ft. flue products.}$$

Since in the average 6,000 cu. ft. of gas are burned, the total volume of the flue products is 242,000 cu. ft. per hour at 440 F. These flue products contain, according to their analysis, a total of 1,680,000 Btu per hour available for heating purposes.

Comfort and Health in Plants Heated with Waste Gases

Movement of air and the humidity are of the utmost importance in factory heating because of the effect on the comfort and health—as well as the efficiency of the workers. In places such as a ceramic plant particles of dust and toxic gases are conspicuously present in the atmosphere and their physical and chemical nature, their concentration in the air and the conditions under which they are breathed determine the harmless or harmful effect of heating with waste gases.

It was mentioned above that the continuous kiln is fired by diffusion burners, especially designed for this particular purpose to produce a slow burning, brightly luminous flame. This delayed combustion is apt to produce (when improperly mixed and controlled) carbon monoxide, which may, of course, occur in any industrial heating process with any kind of fuel. As a result, certain safety standards as to the permissible amount of carbon monoxide which may be formed by gas appliances under extreme operating conditions, have been adopted and have been fully discussed by Dr. F. E. Vandaveer.[1]

The installation described was designed with these factors in mind and utmost care was taken to overcome these probable detriments attributable to heating with waste gases. After the heating plant was in operation, various samples of flue gases taken at different points in the plant were submitted for analysis to the *A.G.A.* testing laboratory which reported as follows:

> "According to our test, this sample contained no carbon monoxide. The carbon dioxide content was very low, only 0.6 per cent being present. This would indicate that the sample was exceedingly diluted, and although our machines are accurate to at least 0.002 per cent, if the carbon monoxide content were less than this amount, we would not be able to determine it."

The Air Duct System

It was calculated that if a sufficiently high pressure head could be built up in a 60-in. diameter manifold, carried 200 ft. alongside the kiln as shown in Fig. 1, this manifold would act as a storage vessel for the waste gases leaving the kiln. The entire plant could then be heated satisfactorily with these waste gases provided, of course, the conventional calculations for a gravity discharge system were followed through with due regard to the various heat requirements of the different buildings. The heat loss of each of the buildings was determined and the available pressure heads established, allowing sufficient margin for friction and radiation losses. An air change every fifteen minutes was thought to be advisable. The observed dry and wet bulb readings were 78.0 and 64.2, respectively, giving 72 F as the effective inside temperature.

The various buildings are vented at the roofs and

[1] *Gas-Age Record* 59 (1927) — 87-90.

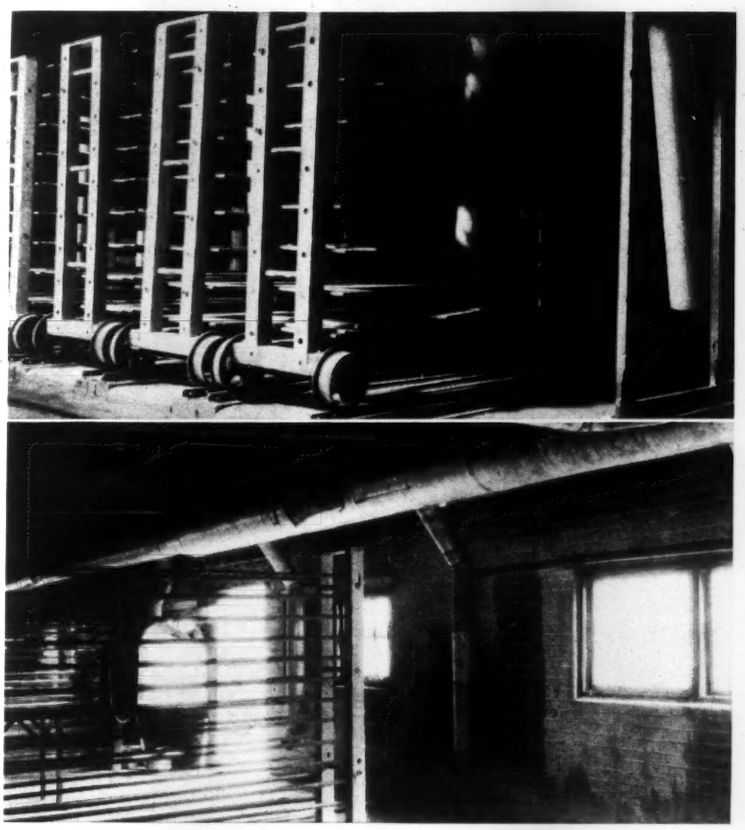


FIG. 3—WASTE HEAT FOR DRYING IS DISCHARGED ABOUT TWO FEET ABOVE THE FLOOR, AS SHOWN IN THE VIEW OF THE DRYERS AT THE TOP. BELOW IS ANOTHER VIEW OF THE DRYERS SHOWING THE DISTRIBUTION AND DISCHARGE LINES

above the windows. In designing the air duct system the friction pressure loss method was used for the calculation and the proportioning of the various mains and branches. The final velocity at the last outlet branch was chosen to be approximately 400 f.p.m. The loss of head in inches of water was figured according to the following formula:

$$h_L = \frac{m\,L\,V^{13/7}}{D^{9/7}}$$

where

h_L = loss of head in inches of water
m = 0.0000000041 for smooth steel pipe
L = length of pipe } both in feet or
D = diameter of pipe } both in inches
V = velocity of air in feet per minute

The ducts are of 18-gage galvanized steel with 4-in. overlap per length, covered with ¼ in. of asbestos paper. Each main branch and each outlet is dampered and provision is made for escape of excess waste heat from the main trunk line into a stack.

There are four large dryers in operation having a capacity of about 400 racks; temperature in these dryers is kept constant at 175 F. The trunk line leading to the dryers is equipped with an automatically-controlled water basin, the humidity in the dryers being controlled by wet and dry bulbs. The gases passing through the trunk line absorb 2.2 gal. of water per hour.

In the event the supply of waste heat is suddenly shut off due to necessary repairs to kiln or car transfer equipment, an auxiliary direct-fired heating system is installed at the end of the main trunk alongside the kiln. This heating equipment consists of a series of automatically-controlled inspirator burners and has a capacity of 1,000,000 Btu per hr. Thermostatic control for the auxiliary heating equipment is located about 600 ft. distant from the source of heat.

Results Have Met Expectations

Results obtained during the last two years with the waste-heat utilization system have been entirely satisfactory. Even in extremely cold weather, an average temperature of 72 F is maintained and the atmospheric conditions in the plant are considered healthy. Because of necessary temporary shut-downs of the tunnel kiln, the auxiliary, direct-fired system has been used a number of times.

Bulletin on Air Flow Through Orifices

"The Flow of Air Through Circular Orifices in Thin Plates" by Joseph A. Polson and Joseph G. Lowther (Bulletin 240 of the Engineering Experiment Station, University of Illinois) describes an investigation made to verify and to extend the range of existing data on the flow of air through square-edged circular orifices in thin plates.

The price of this 40-page bulletin is twenty-five cents, although a limited number are available for free distribution.

Included in the bulletin are a discussion of the equations of flow, a description of the test apparatus, the method of conducting tests, the calculations, a discussion of the results and the conclusions.

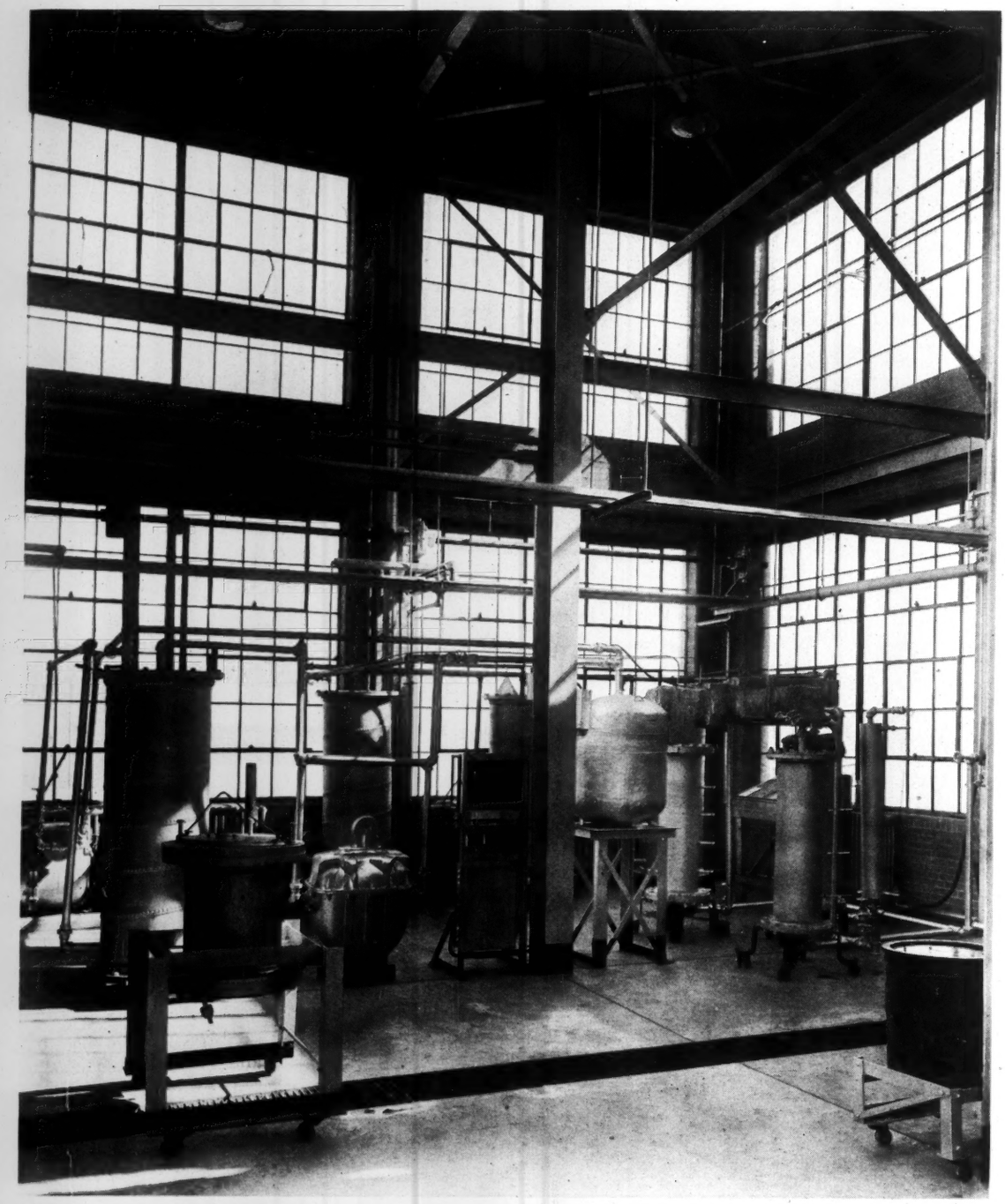


FIG. 1—TWO-STORY-HEIGHT CHEMICAL-ENGINEERING LABORATORY FOR SEMI-PLANT SCALE INVESTIGATION OF PROCESSES

[1] The Aluminum Research Laboratories, opened in January, 1930, are of general interest, not only for their novel features of design, but also because they demonstrate many uses to which aluminum can be put in laboratory construction. Wherever its use was appropriate, economically or artistically, it was employed in the laboratory construction and equipment, and is found to be giving effective service throughout these laboratories. The laboratories have now been in use for two years, so that ample opportunity has been given to demonstrate the practicability and utility of their various features.

The laboratories are located in New Kensington, on a hill somewhat above the prevailing smoke level of the industrial community. Surrounded by green lawns and shrubbery, they offer just a touch of seclusion for the scientist, yet one of the Company's largest fabricating plants is within short walking-distance. The building is 275 ft. long, with 108-ft. wings at each end, with a total of 56,000 sq. ft. of floor space. The three floors are arranged so that the main or center floor is reached by a short flight of steps in front, and the basement floor is at the ground level in the rear. An aluminum elevator in the center of the building is used for carrying heavy equipment and supplies from the basement to the upper floors.

The work of the laboratories is broad in scope. It is divided into divisions, known as the metallurgical, physical testing, analytical chemistry, physical chemistry, chemical development, patent, library, and administrative divisions. With such a varied line of work, the quarters had to be designed to meet a variety of requirements.

Industrial research laboratory buildings present a variety of problems in the design and installation of air conditioning, heating and piping services. Arrangement of the necessary piping systems to provide the requisite flexibility and convenience is of prime importance; ventilation, including the removal of fumes, involves careful study. The Aluminum Research Laboratories, described here, typify the problems met in this type of structure.

Heating, Piping and Ventilating Where Requirements Are Severe

By Junius D. Edwards*

RESEARCH laboratories, as well as research equipment, are seldom standardized. Every new research laboratory is designed with the hope that it will give greater convenience and flexibility of operation than afforded by previous quarters. Sometimes these hopes are disappointed, particularly with respect to ventilation, but continual progress is being made in adapting laboratory buildings to their specific uses.

Ventilation a Major Problem

A major problem in chemical laboratories is the securing of adequate ventilation. Where chemical fume hoods must be operated on many floors, the space required for adequate ventilating ducts is usually very substantial. In the Aluminum Research Laboratories[1] it was found possible to concentrate most of the chemical activities on the third or top floor of the building so that, with one exception, no ventilating ducts had to be provided in the interior or exterior walls. This was accomplished by carrying the exhaust ducts directly from the hoods to the ceiling and through the roof. The problem of ventilating the chemical hoods was further simplified, and flexibility was provided, by combining two hoods, back to back, in a unit, where possible, and employing a separate exhaust fan for each pair of units. In this way the expense of long ventilating ducts with high frictional resistance was eliminated, and the fans do not occupy valuable interior space.

Fans and Motors on Roof

The fan motors are protected by weather-proof steel housings, and with the fans are set on the roof of the building. Each fan and motor are mounted on a steel frame attached to a wooden base which rests without bolting on the roof waterproofing. The roofing tiles are laid up to this base and if necessary would assist in holding the frame in place. The fan runners are of cast iron, protected by paint. Connection between a hood and the roof opening above it is by means of a vertical aluminum pipe 12 in. in diameter. Passing through the roof is a stoneware duct, sealed to make it weather-proof, and connected to the exhaust fan by horizontal sections of stoneware. These horizontal sections are relatively short; not over 6 or 8 ft. in length. Because of the high concentration of acid frequently carried in the fume from the hood, the aluminum exhaust ducts were treated on the inside with one coat of acid-proof paint. An examination of these pipes after more than a year's use showed there had been no serious deterioration of the metal, although the paint (which is not being replaced) had largely disappeared.

Baffle Plate in Hood Distributes Suction

The stone hoods are of a few standard sizes, adapted to their particular use and location; all of them, however, have the suction distributed across the width of the hood by means of a baffle plate at the back of the hood. A suction box at the top of the hood has openings the width of the hood to withdraw air both from behind the baffle plate and in front of it. The construction of the hood in detail is shown in Fig. 3; this design was worked out experimentally to determine the proper size of openings and the air flow required to give the necessary ventilation. Ventilation at bottom of the hood is through an opening in the back baffle plate, which extends to a height of 22 in. from the floor of the hood. Fume which rises to the top of the hood is withdrawn through openings in the suction box placed over the back baffle plate.

Heat Conserved by Drawing Air Through Hoods from Rest of Building

Dampers are provided in the aluminum exhaust ducts and may be used to concentrate the fan capacity on one hood at times. Most of the hoods are concentrated in three large laboratories and by keeping the doors open, the air drawn through the hoods is taken from the rest of the building. This arrangement conserves heat in the laboratories and makes additional radiators in the rooms in which the hoods are located unnecessary. It also pro-

*Assistant Director of Research, Aluminum Company of America, New Kensington, Pa.

[1] See opposite page.

vides some forced ventilation for other corridors and rooms. The fans are selected to pull 500 c.f.m. of air through the standard hood openings, 38″ x 32″. For the single units and for a pair of units, 1-hp. motors are used to operate the exhausters. In a few cases, three units are connected to the same fan, and in these cases a 3-hp. motor is employed. With this design, it has been found unnecessary to provide doors on any of the hoods to prevent fume from spreading into the laboratory. The fan motors are equipped with remote control operated by buttons with signal lights placed conveniently near each hood.

The hoods, in most cases, are located at the ends of the chemical benches near the center of the room and are arranged in units with two hoods back to back. In some cases, however, there are individual hoods placed against the wall. Each chemist is provided with a laboratory bench with a sink at the end near the wall and a fume hood at the other end. He also has a narrow work table and a writing table with chair. The freedom from fume in the chemical laboratories is such that analytical balances can be safely used in alcoves in the laboratory without being confined behind closed doors, a decided convenience.

Metal-Covered Pipe Chases on Outside Walls

Another problem in laboratory design is the piping systems. Water, steam, gas, electricity, and compressed air must be provided in nearly all the laboratories, as well as must suitable drains to carry off a variety of chemical wastes. From the standpoint of appearance, it is undesirable to carry the piping along the ceilings of rooms and corridors, and from the standpoint of convenience for inspection, change and repair, the piping should not be buried in floors or inaccessible ducts. The plan adopted was in part a compromise, for the water, steam and gas mains are carried on the corridor ceilings in the basement. These mains feed into vertical pipe chases which are placed at every outside column of the building. Every laboratory or office has one or more of these metal-covered pipe chases on its outside wall, through which the various services can be piped. The shafts with their removable metal covers are about 35½″ x 13½″ in section and, being finished to match the metal partitions, are not obtrusive in size or appearance. One serious objection to carrying piping through the floors is that liquids spilled on the floor around the pipes will leak through to damage the ceiling below or whatever the liquid may fall on. The pipe chases in these laboratories come through the floors, but are protected from leakage by a steel plate set in the concrete. This steel

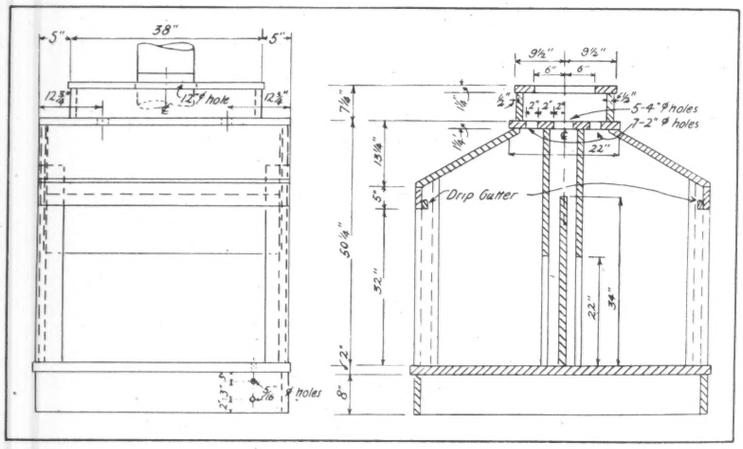
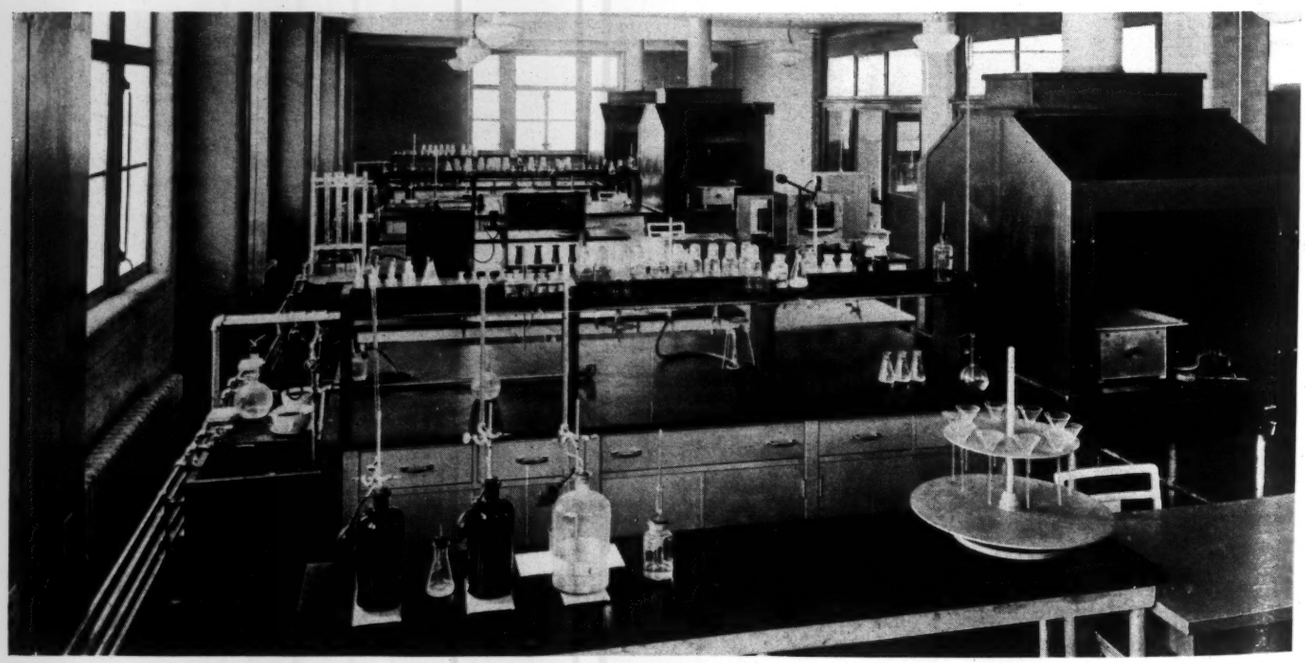


Fig. 2 (Below) — The Chemical Laboratory Showing the Fume Hoods. Note the Pipe Chases on the Outside Wall

Fig. 3 (Right)—Detail of Fume Hood



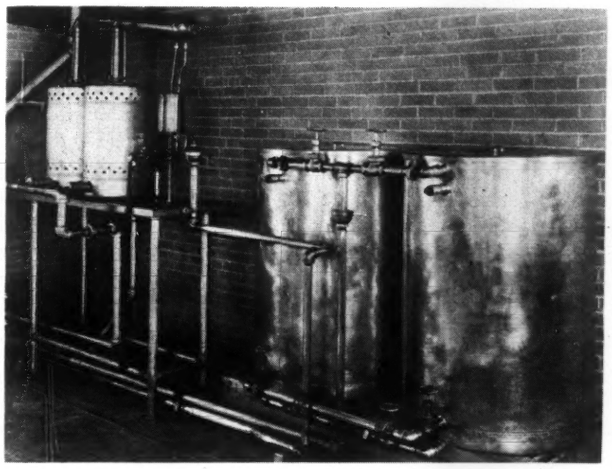


FIG. 4—ALUMINUM STILLS FOR THE PRODUCTION OF DISTILLED WATER

plate has a flange projecting above the floor level which provides anchorage for the pipe chase cover and prevents leakage into the pipe chase. The laboratory and corridor ceilings on the second and third floors are free from piping which adds considerably to their appearance.

Drain Pipes Arranged for Easy Cleaning

By placing laboratory sinks next to the outside wall, the average pipe carry from chase to laboratory sink and bench is only a few feet. This is particularly important in the case of drain pipes, which in the chemical laboratories are constructed of chemical stoneware. In case a drain should clog up, an opening is provided on the roof which permits passing a ram straight through to the basement. At the bottom an alloy-metal opening is also provided for cleaning out the horizontal section, which in each case leads directly to a manhole in the main sewer placed outside the building. All the laboratory piping is arranged by this system to be carried along the walls of the building and at a low level, below the windows, where it is readily accessible. Doors and walls in the halls and partitions do not interfere with the piping, and the removable steel partitions can be changed without disturbing the piping. Partition changes were required within a year of occupancy and were readily made.

The laboratory benches are of aluminum with stone tops and mop boards; drawer pulls and other fittings are of aluminum alloys which are not discolored by hydrogen sulphide.

Distilled Water Conveyed in Aluminum Piping

The water stills are placed in the pent-house and the distilled water is collected in two 125-gal. aluminum storage tanks. From these tanks it is distributed by gravity through aluminum piping to outlets placed in the main laboratories. Because of the low yield point of aluminum piping, care should be exercised in assembly to pull fittings up to a snug fit, with less pressure than is used with iron pipe.

When water is first passed through such a system, a protective oxide coating is formed on the interior of the aluminum piping; some aluminum passes into solution during the formation period. Passing tap water through the system for a day or two will clean out the piping and promote the formation of the protective coating and with a week's use of the system, it should be satisfactory. Laboratory determinations have shown less than 1 part of aluminum in 100 million parts of distilled water handled in this way.

Electric Power Purchased

Electric power for the laboratories is purchased. From the transformer and motor generator room in the boiler-house, it is distributed to switchboards in closets at the ends of the halls. Conduit, in the basement floor between all pipe chases, and in other floors between the switchboards and pipe chases, permits the convenient distribution of electric current to the laboratories and makes additional outlets easy to arrange. Current for the lighting systems is distributed through conduits laid in the concrete floors.

Vapor-Vacuum Heating System Used

The heating plant is placed in a two-storied section of the building, located in the central court. The boilers are double-pass fire tube boilers, gas-fired with automatic control. They are connected to a 135-ft. stack, and so arranged that they can be converted to burn coal with a minimum of change and expense.

A vapor-vacuum system is employed in the heating of the laboratories; two vacuum pumps with automatic control are used in operating the system. In the offices, the radiators are concealed by ornamental aluminum covers, open at the bottom and with an aluminum grille at the top. Radiators themselves consist of a length of aluminum tubing bent back and forth horizontally to form a flat coil, to which is tightly fitted a series of vertical aluminum fins.

Double-Storied Laboratory for Tall Equipment

Fig. 1 shows a section of a double-storied laboratory located in the central court used for chemical process development work; the extra height permits the erection of tall equipment. Most of the apparatus used in this laboratory is set up temporarily for some particular investigation, and the piping and other equipment are arranged to give convenience and flexibility of operation. The apparatus shown in the cut is being used in a study of the absorption efficiency of activated alumina, a special form of alumina capable of absorbing up to about 15 per cent of its weight of water. It is used in commercial air-conditioning; heating of the saturated activated alumina at relatively low temperatures regenerates it for use in another cycle of moisture absorption.

Aluminum Paint Used on Piping

In this laboratory, as well as in the rest of the building, the piping is protected by aluminum paint, which in addition to its protective properties offers an advantage in its silvery surface which reflects light freely, adding to the lighting efficiency.

Economy in Drying

Aided by Vapor Proofing

By Malcolm Tomlinson*

A DEFINITE barrier erected against moisture flow through walls, ceilings and floors will aid the economical and satisfactory operation of drying rooms, laboratories where humidity control is used, storage rooms where the moisture content of the atmosphere must be kept at a low level, rooms that are dehumidified and cabinets such as incubators for animal industry investigations, bacteria breeding and the hatching of poultry as well as brooders for baby chicks, etc. Humidified compartments can often be sealed to advantage against the outward flow of moisture.

The natural tendency of water vapor is to seek equilibrium. If the absolute humidity is not identical inside and outside and there is any way for this vapor to move back and forth the speed with which equilibrium is established will depend upon the humidity difference and the difficulty encountered due to intervening membrane. Since practically all building materials are hygroscopic in structure so that adsorption and absorption of moisture are functions of the temperature and the relative humidity some sort of proofing compound must be applied to one or both surfaces if vapor flow is to be prevented. Such compounds must have sufficient elasticity and strength to prevent rupture from settlement of the building and the pressure exerted, naturally or artificially, by the inside or outside atmosphere. In addition it must have a finish such as to insure that its own pores are closed.

Moisture equilibrium may be established through open doors and windows, through cracks and through porous or hygroscopic membrane in materials of construction. The amount which flows by the way of doors and windows can be kept at a minimum by means of air locks and corridors similar to those used in cold-storage rooms. Furthermore such leakage can easily be calculated and is not continuous. The water vapor which enters or leaves with the air through cracks is a continuous process not easily estimated because it is difficult to determine the average size of the cracks. Since such leakage goes on continuously it is costly but it can be stopped.

The vapor which enters or leaves through the walls, ceilings and floors of a room is the most insidious flow of the three mentioned because it is so minute per sq. ft. of exposed surface and it usually finds no barrier against its flow. The total amount of moisture flowing through membrane may be sufficient to interfere with production or raise costs excessively. Superficially it might seem that the moisture flowing by the three means named is of little importance. However, when it is considered that one of the largest losses in heating is the infiltration of cold air through door and window cracks and that this loss is often 20 per cent of the total heat available from the fuel used, failure to consider the vapor flow factor seems to be of some importance.

Little research data are available on the flow of moisture through membrane. Nevertheless there are available much data which, if used, will give valuable information for practical applications. One valuable case may be mentioned. In engineering a ½-per cent relative humidity room, 12-in. brick walls, a 12-in. tile-and-cement ceiling and a 12-in. concrete floor were used. The surrounding space had a variation in relative humidity between 30 and 50 per cent, a maximum temperature of 90 F and gravity air motion. Within, the temperature was 125 F and the air motion 100 f.p.m. Precautions were taken aganist vapor entrance by doors and through cracks but the necessity for vapor proofing was not realized. After the equipment was in operation it was found impossible to obtain less than from 2 to 3 per cent relative humidity until the walls and ceiling had been water-proofed, when the relative humidity immediately fell to ½ per cent. In this case the vapor pressure outside the room varied from 0.21 to 0.56 lb. per sq. in., the vapor pressure within the room was 0.04 to 0.06 under highest temperature conditions and the desired vapor pressure was 0.01. Since the space enclosed was 65,000 cu. ft., vapor-proofing stopped the inward flow of from 5½ to 9¼ lb. of moisture per hour.

"But what of that," some one will say, "such conditions do not occur in actual practice." As a matter of fact even greater differences are often had but the factor of safety provided by equipment takes care of the extra work involved. Thus the engineer often does not realize that his hourly costs are higher than they need be.

It is interesting to note that a ratio of from 1 to 5¼ between the inside and outside vapor pressure gave a flow of 5½ lb. of moisture while a ratio of 1 to 9¼ gave a flow of 9¼ lb. The increase per point in the ratio thus represented an increase of one lb. of water. The vapor proofing which effectively stopped moisture flow in this case, under a maximum vapor pressure difference of 0.55 lb. per sq. in., consisted of two coats of zinc paint covered by an egg shell finish for walls and ceiling. This proofing was afterward superseded by aluminum foil which, for walls and floors, was covered by rag felt saturated with asphalt and, for ceilings, was coated with varnish.

The structural differences between the many materials used for building construction and the temperatures required will naturally call for variations in the type of vapor proofing applied. Nevertheless, whether the membrane to be covered be a cellulose, a mineral or an animal product its pores must be completely closed with a water- or a vapor-proofing compound which, when dry, will have the strength to resist the possible vapor pressures and which will insure that all pores are closed.

*Consulting engineer, Urbana, Ill.

What the Hartford Loop Is And What It Does

By Robert E. Hattis*

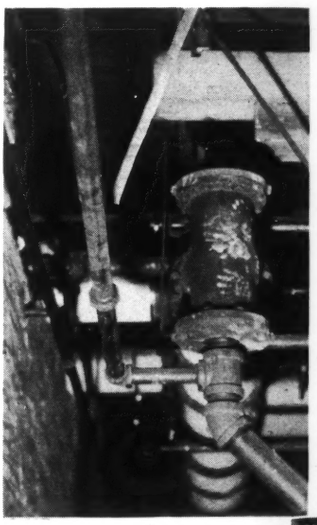
There is considerable misunderstanding regarding the use of the Hartford loop, what its intended function is, and how it is applied. This article aims to bring opposing views to closer agreement by a clear explanation of the loop and its use.

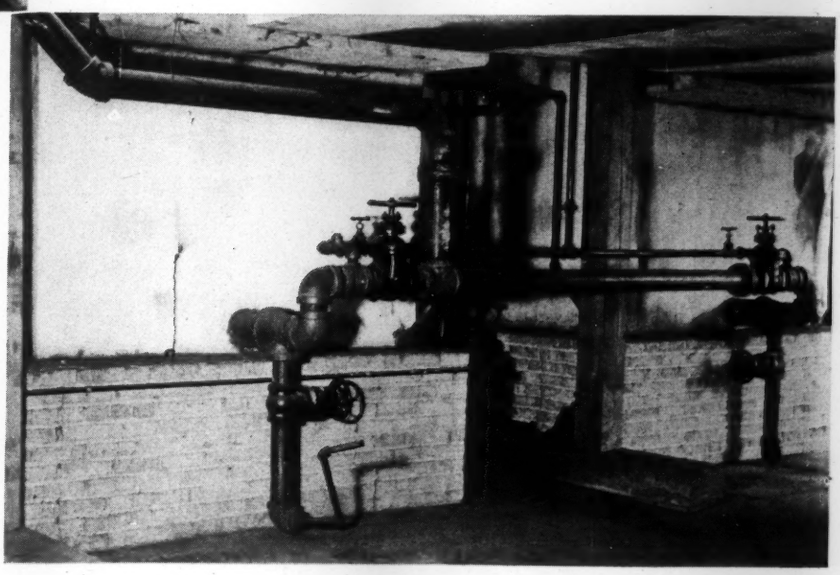
THE intention of this article is to explain what the Hartford loop is, what it does and what piping is involved in its use on various boiler installations. It is thought that by a clear and thorough explanation of the loop as actually applied, much of the difference of opinion which the subject has aroused can be brought together a little more closely.

What the Loop Is

Fig. 1, taken from a leaflet issued by the Hartford Steam Boiler Inspection and Insurance Co. ("The Hartford Water Line Return Connection for Heating Boilers") illustrates the loop connections in single- and multiple-boiler installations. Essentially, the loop consists of a line connecting the steam and return sides of the boilers without any intervening valves, into the vertical run of which is connected the heating system return line at the lowest safe operating water level. The Hartford company recommends that the top of this horizontal pipe connection shall not be lower than the lowest permissible operating water level, usually the bottom of the water-gage glass. However, as water can leave the boiler until the water level reaches the bottom of this horizontal pipe, this bottom must be no lower than 1 in. above the top of the fusible plug in the boiler. The return pipe and pipe *B* should be of the size prescribed by modern practice for return lines. The recommended size for pipe *C* is:

Grate Area	Size of Pipe *C*
4 sq. ft. or less	1½ in.
4 sq. ft. to 15 sq. ft.	2½ in.
15 sq. ft. or more	4 in.





Adapted Hartford Loop Installed in Connection with Two Oil-Fired Heating Boilers in a Loft Building. To the Right Is a Rear View of the Boilers Showing the Return Line. The Horizontal Pipe at the Top of the Picture Above Comes From the Vacuum Pump

* Consulting engineer, Chicago, Illinois.

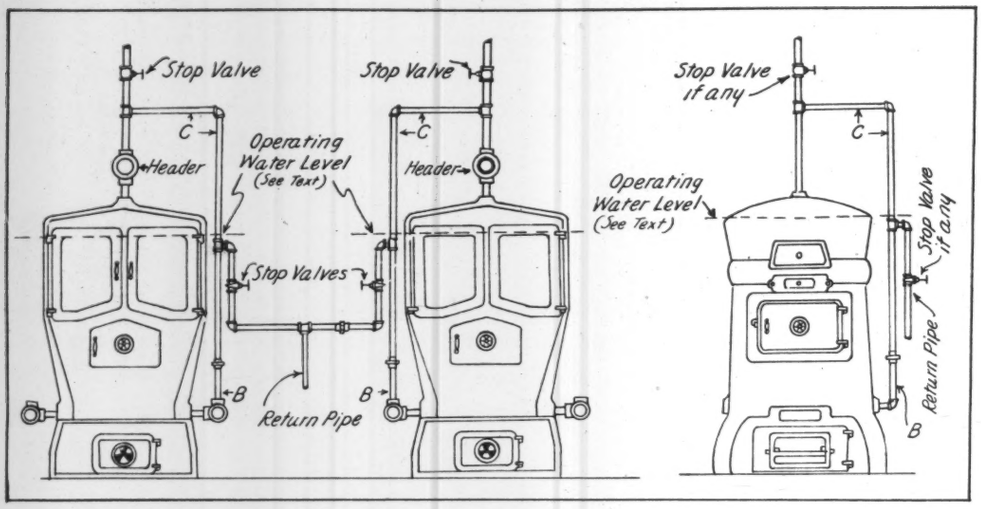


FIG. 1—LEFT—METHOD OF CONNECTING TWO OR MORE BOILERS. RIGHT — METHOD OF CONNECTING RETURNS TO A SINGLE BOILER

What the Loop Does

Boiler accidents may occur as a result of low water; cracked sections or overheated plates and tubes will cause shut-downs, inconvenience, expensive repairs and sometimes endanger life. The Hartford loop needs no attention as it has no moving parts to be maintained, prevents the loss of boiler water from below the danger line, and in its operation on a low-water condition causes a water hammer which calls attention to that fact. In most cases it costs little to install; the loop was designed and offered without patent by the Hartford company for free use by the heating industry.

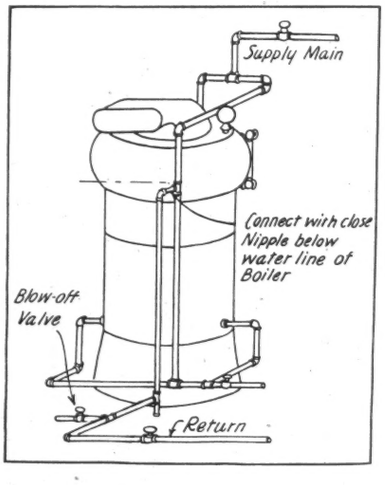
It may be of interest to state here some facts obtained from E. R. Fish, chief engineer of the boiler division of the Hartford company, relating to low-water accidents. Each of the following causes for low water has been responsible for accidents:

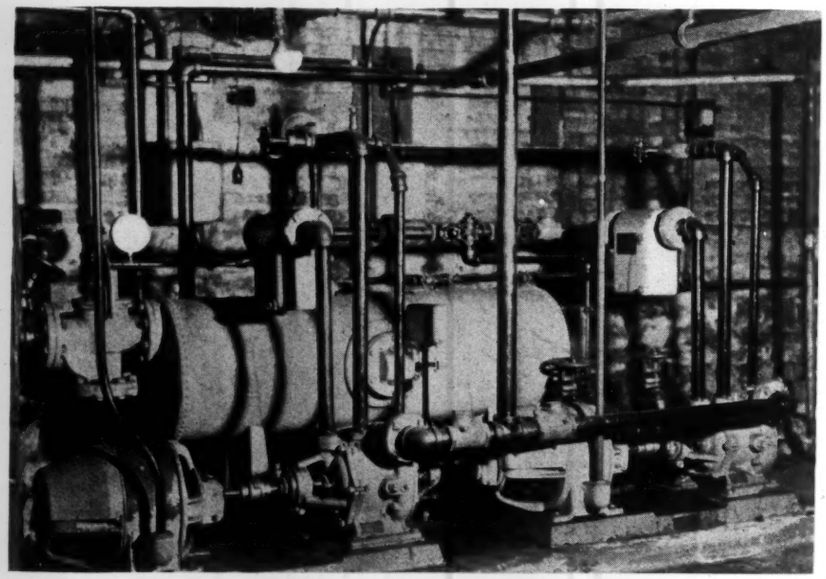
Improperly designed or installed piping may make it possible for considerable quantities of water to be held in the system and not returned promptly to the boiler. The need for hot water may prompt some one ignorant of the possible consequences to draw it from a conveniently located drain valve, without making an effort to add an equivalent amount.

Leakage may occur from a return line which is not discovered until the boiler is short of water and damage done.

Use of a heating boiler as a refuse burner in the summer may raise steam, the presence of which in the radiators is objectionable. Consequently, the valve in the steam line is closed. As soon as pressure is raised, the water backs out of the boiler if

FIG. 2—HARTFORD LOOP ON ROUND CAST-IRON BOILER





THE PUMPS WHICH RETURN CONDENSATE TO THE BOILERS SHOWN ON THE PRECEDING PAGE. THE VERTICAL LINE IN THE CENTER GOES TO THE BOILERS. NOTE THAT THE PUMPS ARE INSTALLED IN DUPLICATE

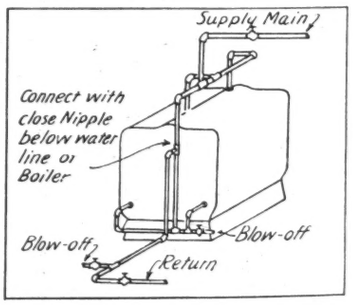


FIG. 3—HARTFORD LOOP ON SQUARE CAST-IRON BOILER

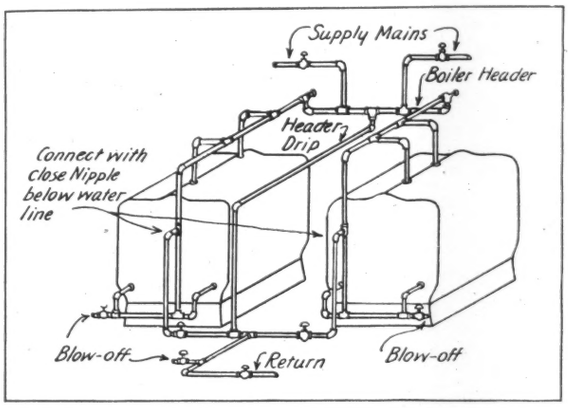


FIG. 4—HARTFORD LOOP ON TWO SQUARE CAST-IRON BOILERS IN PARALLEL

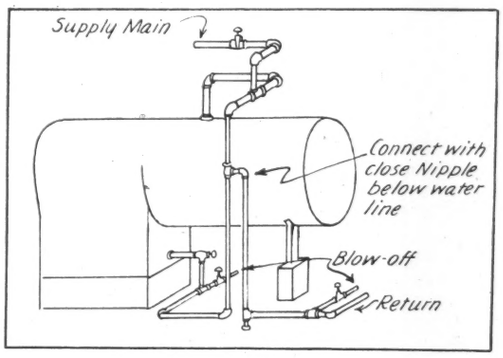


FIG. 5—HARTFORD LOOP APPLIED TO A STEEL BOILER

there is no check valve or if the return line valve is left open. The same thing may occur where the heating boiler is used to heat water in the summer and gets up pressure.

A water gage glass connection may become plugged, trapping water in the glass and giving a false indication of the water line.

Improperly manipulated steam and return-line valves.

How It Prevents Low Water

How the loop prevents low water is evident from Fig. 1. It can be seen that closing the valve on the steam line cannot cause the loss of water below the bottom of the loop horizontal pipe connection. This is also true in the case of leaks in the return lines. However, the loop cannot prevent the loss of boiler water by evaporation, if for any reason the condensate should fail to be returned. When the water line drops low enough to admit steam into the horizontal loop connection, water hammer occurs, calling attention to the trouble. As this water hammer does not occur until the water line drops to a dangerous level, the noise feature seems to the writer an asset, although some heating engineers find fault with the loop because of this. As in many boilers the bottom of the gage glass is also the lowest safe water level, the top of the loop horizontal connection is higher than this by the diameter of this pipe, which will cause water hammer before the danger line is reached. In this case it is recommended that a 2-ft. high air chamber be installed in the return line near the loop connection to absorb any shock.

Its Use with Boilers in Parallel

In addition to preventing low water the loop seems to have some other advantages. One of these is in connection with the parallel operation of boilers, where the loop acts to equalize the water levels as long as it is submerged. However, it is possible even with a submerged loop to have different water levels in the boilers operating in parallel, if different steam pressures prevail in each boiler. In this case the lower water line will exist in the boiler with the higher pressure. Another advantage is claimed for the loop by some heating engineers who report a tendency for lessened boiler corrosion from air returned with the condensate. The entrained air released in the loop rises to the boiler steam space in the equalizing pipe, instead of rising up through the boiler.

While the loop was designed primarily for use on gravity-return systems, the question often arises as to its desirability on pump-return systems. With return pumps the delivery of condensate to the boiler is more positive than with gravity systems, but with the possibility of interference of some sort that may result in low water, the noise alarm feature of the loop seems to be desirable. Furthermore, since pump returns are smaller than gravity returns, failure of a pump due to current or other interruption may result in failure to pass the condensate back to the boiler fast enough when the system is operated on a by-pass. The resulting low water will again be announced by water hammer.

As it is sometimes difficult on return connections to control the amount of water returned to each of a battery of boilers operating in parallel, the submerged loop connection making for water level equalization offers some advantages here too.

The Hartford loop is evidently a desirable return connection and if the piping involved in its installation does

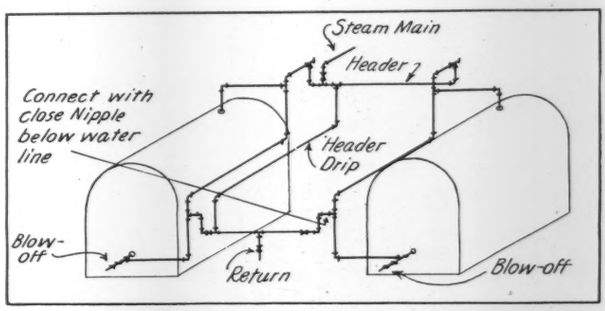


FIG. 6—HARTFORD LOOP ON TWO STEEL BOILERS IN PARALLEL

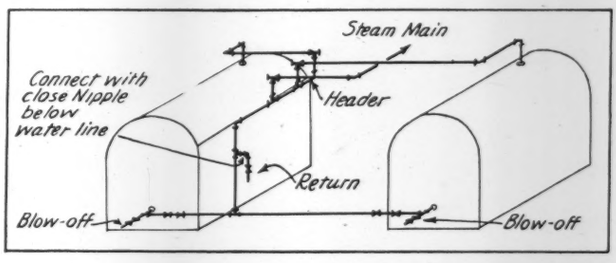


FIG. 7—ADAPTED LOOP ON TWO STEEL BOILERS IN PARALLEL, SHOWING REDUCTION IN PIPING OVER INSTALLATION ILLUSTRATED IN FIG. 6

not become too complicated and expensive, its use is often advisable. The sketches herewith show single- and multiple-installations of various types of boilers showing just what piping is involved, facilitating a decision as to its desirability. The writer's practice as to its use are as follows:

Type of Boiler	Installation	Remarks
Cast Iron	Single	Loop recommended.
Cast Iron	Multiple	Loop recommended.
Steel	Single	Loop recommended.
Steel	Multiple	Adapted loop recommended.

Effect of Relative Humidity on Cellophane

THERE are two types of Cellophane, one the moisture-proof or special process stock and the other the non-moisture-proof, which is a plain regenerated cellulose sheeting containing glycerine for holding the proper moisture and keeping the film soft and flexible.

The writer has seen Cellophane work satisfactorily in humidities as high as 80-85 per cent but does not consider storing Cellophane under such conditions advisable; neither is a humidity as high as 66 per cent for storage purposes advisable. Absorption of moisture by the film may cause a sticking and wrinkling condition which is objectionable, not only from the standpoint of machine operation but hand application as well.

Temperature has no effect upon Cellophane except to cause it to lose moisture or absorb moisture more readily, depending upon the humidity. The absolute moisture in the air remaining constant, temperature changes from 45 to 100 F should have no effect on the Cellophane.

In winter, Cellophane has a tendency to lose moisture as a result of dry weather and become somewhat brittle. In summer it has a tendency to absorb small amounts of moisture and stick slightly. These difficulties, however, have been largely removed by the improvement made in the quality of Cellophane during the past few years.

We store Cellophane under optimal conditions if possible, which is 50 per cent relative humidity. Extremes of temperature and humidity should be avoided.—*R. C. Schilly,* Du Pont Cellophane Company, Inc.

Use of Cellophane in a Cigar Factory

Use of transparent cellulose sheeting requires rather close supervision in several ways. Variation between the various brands of this type of wrapping material which are now on the market necessitates inspection of incoming shipments.

We find that relative humidities of 60 per cent to 66 per cent are satisfactory as long as our control will maintain an even condition within this range. The temperature, however, must not be allowed to exceed 80 F. Over this point the sheeting becomes decidedly "gummy" which, of course, makes it difficult for a high speed wrapping machine to operate smoothly.

When electric heaters are used for sealing purposes the heat output and the effect this may have on the maintenance of suitable air conditions should be considered.—*R. D. Touton,* Bayuk Cigars, Inc.

Methods for Sampling and Analyzing Steam and Condensate in Corrosion Studies

The corrosion committee of the *National District Heating Association* (a sub-committee of the research committee) has recently prepared a 20-page booklet entitled "Standard Methods for Sampling and Analyzing Steam and Condensate in Corrosion Test Studies." While copies of the booklet are not available for general distribution, according to D. L. Gaskill, secretary of the Association, a number were prepared for distribution at the *N. D. H. A.* meeting.

Causes of Corrosion

According to the introductory paragraphs of the pamphlet "Corrosion in steam and condensate lines is due principally to the presence in solution of two gases, oxygen and carbon dioxide. They are introduced either through the medium of the boiler feedwater, in which they may exist either as free or chemically combined gases and are liberated by heat in the boiler, or by inleakage of the atmosphere extraneous to those parts of heating systems which periodically or continuously operate at sub-atmospheric pressures.

"Analyses of the concentration of these gases in a sample of steam which has been intentionally condensed or in condensate produced in a building heating system are frequently required in corrosion studies. Unless such samples are collected and analyzed with due attention to certain restricting principles, the results obtained may frequently lead to erroneous conclusions. In order, therefore, that the results obtained by different investigators may be compared and to eliminate faulty methods, it is desirable that certain standards for sampling and analytical procedures be established."

The booklet then describes and illustrates set-ups for taking samples under various conditions and gives detailed information on analytical procedure.

A Half-Century of District Steam Service Described

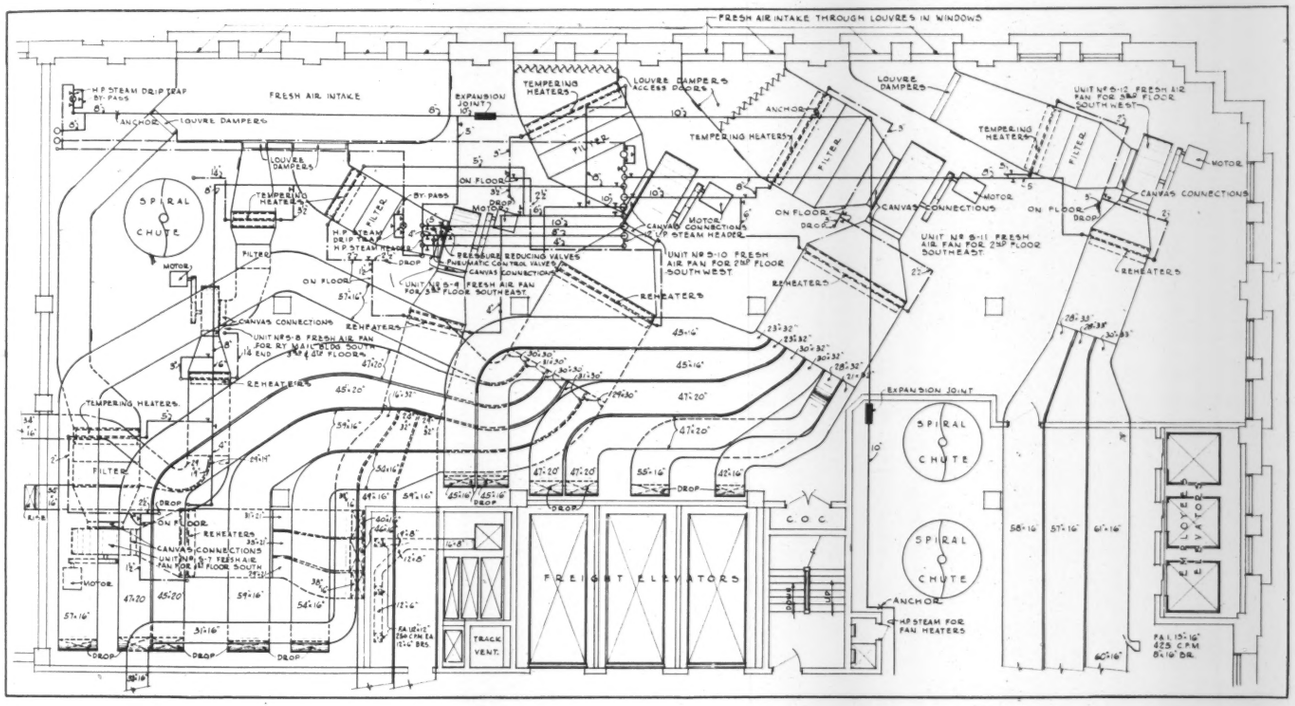
Fifty years of district steam service in New York are commemorated and described in a 136-page anniversary book recently published by the New York Steam Corporation. Starting in 1879, life in New York City is described and the facts about the founding and development of district steam service are included as a part of the general historical survey. Numerous illustrations, including old prints, photographs and drawings prepared especially for the book, portray in a striking manner the growth of New York and the improvements which have been made in district heating.

The book makes fascinating reading, describing as it does the difficulties encountered in laying the first mains, the search for a proper method of metering steam used by customers, insulation difficulties, etc. The concluding pages of the book are devoted to a description of the Corporation's plants today, its growth, earnings, etc.

Ventilating System Designed for Efficient Operation

By George W. Ott*

PLAN OF THIRD-FLOOR FAN ROOM, SOUTHEAST CORNER (16'0" CLEAR CEILING HEIGHT). MAIL CHUTES, INSPECTORS' PASSAGES, ETC., COMPLICATED THE DUCT LAYOUT



VENTILATION is a very important feature in the new Chicago post office,[1] in order that the employes can work at the highest efficiency and speed the mail to the destinations in the shortest time possible. Natural ventilation obtained from windows will be supplemental with a mechanical system of fresh air supply and exhaust, because the areas involved will be very large.

Building Is Over Railroad Tracks; Removing Smoke from Locomotives

The entire area which will be below the building is occupied by the Chicago Union Station for tracks and platforms which must be kept clear of smoke and fumes so that no discomfort will be experienced by passengers arriving or leaving, or by employes working in this area. To keep the air clean and wholesome, a suspended concrete ceiling will be installed over the entire area at the clearance line; between the top of this ceiling and the first floor will be a plenum space which will be entirely enclosed except for long smoke troughs placed above and running full length of each track. Smoke and fumes which are discharged by locomotives will be delivered into this plenum chamber, where they will expand and be drawn out of the chamber through eight concrete ducts uniformly distributed and connected to exhaust fans at the top of the building.

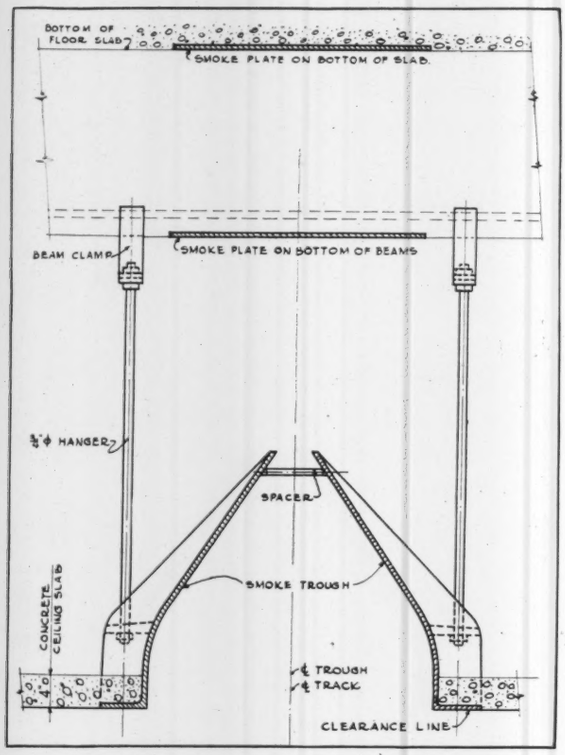
Sizing such a plenum for locomotive smoke removal is a matter of using all of the space that is available. The larger the plenum chamber the more effectively the smoke is removed.

The exhaust fans will discharge the fumes and gases to atmosphere. Practically the only opening into the track space being at the south end of the building, the exhaust fans will create a constant flow of air into the track space and thence into the plenum chamber. Two fans will be installed at the top of each of six of the shafts, one or both of which may be used as the need arises. Single fans are to be provided at the top of the other shafts.

Total capacity of the fans will be 560,000 c.f.m., the largest having a capacity of 94,000 and the smallest 22,500 c.f.m. They will be of extra heavy construction throughout and painted with two coats of asphalt paint to prevent the corrosive action of smoke and gases; ducts joining the mains to the fans will be extra heavy

* Mechanical department, Graham, Anderson, Probst & White, architects, Chicago, Ill.

[1] The general features and the heating of this building were described last month.



SECTION THROUGH SMOKE TROUGH OVER TRACKS, INDICATING METHOD OF LOCOMOTIVE SMOKE REMOVAL

gage metal painted with two coats of black paint to protect them from smoke.

Lobby Air Circulated Through Booster Heaters

In addition to the 14 fans used for the smoke exhaust system, there will be 64 other fans having a total capacity to exhaust 1,573,000 c.f.m. used for the general exhaust system in the building. Fresh air will be provided by 43 fans having a total capacity of 1,170,025 c.f.m. These will be supplemented by 12 fans used for heating the public lobby, taking air from points near the ceiling and recirculating it through booster heaters and discharging it at the sides of the revolving doors near the floor; this system will be augmented by 5500 c.f.m. of heated fresh air and 3140 sq. ft. of direct hot-water radiation.

In order to reduce the operating cost, the fresh air and exhaust systems are so designed that it will be possible to shut off various parts of the building when they will not be in use, or to shut off portions of the system when the number of men working in these portions is reduced and the ventilation can be reduced without causing a decrease in efficiency.

Air Exhausted at Floor or Ceiling to Promote Comfortable Working Conditions

All work spaces will be provided with fresh air supply and exhaust except portions of the trucking areas and loading platforms in the first story where exhaust only will be provided. In the trucking areas the exhaust is figured to change the air in the space ten times per hour; when fresh air is supplied in the trucking areas an amount equal to the exhaust will be furnished. Sixty per cent of the exhaust will be taken out through the face of the platform and forty per cent at the ceiling directly above the platform, thereby removing the gases from the trucks before they can spread over the areas where men are working. In the work spaces the air will be supplied at the ceiling and the exhaust will be arranged to be taken out at the ceiling in the summer and at the floor in the winter, or part at each place as the conditions warrant; this will tend to make conditions at the working level more comfortable.

In the office portions of the first and second stories six changes per hour will be supplied and exhausted; in the work spaces from the third to the eighth stories four changes of air per hour will be supplied and exhausted, while in the empty equipment space on the ninth story five changes per hour will be exhausted, while only four changes will be supplied per hour in order to prevent dust from the empty equipment working space getting into the surrounding spaces.

1500 Tons of Ducts

Approximately 1500 tons of metal will be used for the ducts. Exhaust ducts, located in the plenum chamber over the tracks, exhausting air from the face of the loading platforms in the first story and ducts exhausting from the plenum chamber will be of a special composition, to resist the corrosive action of the smoke and fumes. Ducts exhausting from the battery charging room and exhausting from the chemical hoods in the laboratories will be of acid-resisting metal. Kitchen exhaust ducts subject to fire, and the discharge duct from the kitchen exhaust fan, will be of No. 14 gage black iron. All other duct work will be of galvanized iron of the following gages:

Ducts 24 in. and less in width or depth No. 24
Ducts 25 in. to 36 in. in width or depth No. 22
Ducts 37 in. to 60 in. in width or depth No. 20
Ducts 61 in. or over in width or depth No. 18

Fan chambers and fresh air intakes will be of No. 12 gage metal. All duct work, fans except portions exposed to smoke and fumes, heater casings, motors, frames and louvred dampers, will be painted with one coat of black metallic paint suitable for hot surfaces, and all such iron work which will not be covered will be finished with one additional coat of paint. The portions of the fans that are exposed to the smoke and fumes will be painted with two coats of bitumastic paint. Where covering will be installed it will be painted with two coats of asbestos paint of a light color. Metal portions of fresh air intakes will be covered with 1½-in. asbestos air-cell blocks, installed in two layers with joints broken, and covered with 8-oz. canvas, painted.

Heater Stacks Pneumatically Controlled

The temperature of the air entering the rooms varies from 70 to 80 F, depending on whether the room has an outside exposure or is entirely an inside room. Heaters will be arranged in two groups of two stacks each (tempering and reheaters), all heaters in both groups being controlled by pneumatic control valves.

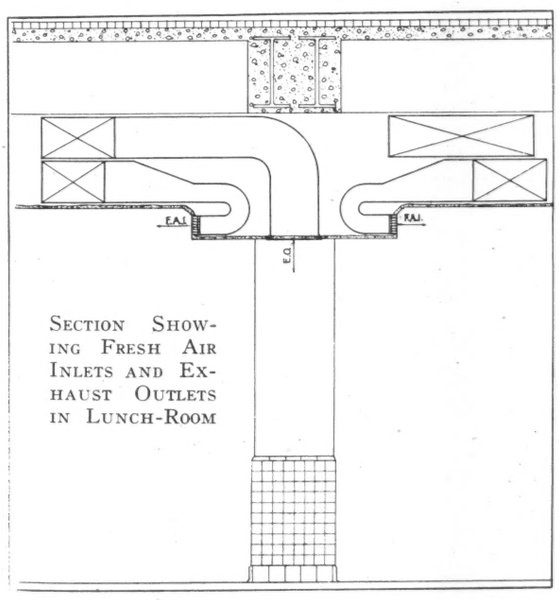
Automatic oil filters of the self-cleaning type will be used for cleaning the air. Hand-operated dampers which can be secured in any position desired will be provided in the fresh air intake of each fresh air fan.

All fan heaters will be of seamless copper or brass tubing, having a helically-wound copper or brass extended surface on the outside surface. Extended surfaces will have a uniform pitch, extended at right angles to tubes and will be diametrically integral with the tubes. Heaters will be designed for a maximum pressure of 150 lb. per sq. in. The tubes will be arranged for three passes and will be provided with unions at each end so that any tube may be easily removed. Each section will have a suitable galvanized steel casing of ample strength for supporting tubes and arranged for connecting to the casing built up for each fan unit. Supply and return connections will be on opposite ends. The velocity through the net area will be approximately 1000 f.p.m. The heaters will be supported from the floor by substantial metal stands on structural steel framework. All fan heaters, except those for warming the public lobby, will be connected to and made a part of the steam-heating system; fan heaters for warming the public lobby will be connected to and made a part of the hot-water heating system.

Fans Have 6% Excess Capacity for Leakage

The kitchen fan will be equipped with a fire damper, counter-weighted, with a fusible link, which will be arranged to close against the inlet of the fan and by-pass the flames in case of fire, so that the fan will not be damaged. A hand-operated damper will be provided in the discharge of each exhaust fan. The general exhaust is discharged to the atmosphere not lower than the third story, but most of the fans are located above or at the main roof level. Fumes from the track level, kitchen and chemical laboratories are discharged at or above the main roof level.

Owing to space conditions, the fans will all be built into close spaces. The fans are all designed to operate quietly and without vibration at all speeds and to have an air capacity not less than 6 per cent greater than that needed as supply or exhaust for the rooms served. This added amount is to compensate for the leakage which may occur between the fans and the rooms served. Fans for chemical hoods in the ninth story will be constructed of an acid-resisting material. Fans and motors will be erected on two inches of machinery cork to absorb the vibration and noise.



Section Showing Fresh Air Inlets and Exhaust Outlets in Lunch-Room

Induction Motors Drive Fans with V-Belts

Short center V-belt drives will be used between the fans and motors. Some of the fans will be located where it is necessary to install double inlet, double width fans, in which case the shafts will be extended outside of the housings. In these cases a pedestal-type bearing will be located between the drive pulley and the housing. Heavy double canvas connections will be installed on the inlet and discharge connections of each fan connected to a steel duct to prevent transmitting vibration of fan to duct system, except for the kitchen exhaust fan which will have asbestos cloth connections.

Fan motors will be of the squirrel cage induction type wound for 220-volt, 3-phase, 60-cycle alternating current. The fan horsepowers will be based on a fan efficiency of 50 per cent and will be designed to operate at constant speeds. All motors will have sliding bases.

Grilles in marble walls will be of cast bronze, of a special design built up in layers with the air discharging around the edges of the built-up layers. In plastered walls, furred beams and ceilings, the fresh air inlets will be stamped steel plain lattice pattern, with a free area of 67 per cent, complete with lock louvres. Fresh air inlets in the work areas will have beaded outlets, while exhaust outlets in these spaces will have beaded outlets with wire screen. Behind all inlets and outlets will be placed diffusers to spread the flow of air over the entire face of the openings. Fresh air will be supplied through the grilles or openings at a velocity of 300 f.p.m., while the exhaust air will be removed through the exhaust grilles or openings at a velocity of 350 f.p.m.

Complete System of Temperature Control

A complete system of temperature regulation will be installed for the hot-water heating converters, domestic water heaters, tempering and reheating fan heaters and mixing dampers. Each section in each group of heaters will be individually controlled, as will be the mixing dampers. In the fresh air intake of each fan there will be installed a duct-type thermostat to control the first section of tempering heaters and to keep the steam to the heaters on at all temperatures below 55 F. The second section of tempering heaters will be controlled by a duct-type thermostat in the tempered air chamber. The reheaters will be controlled by a two-point thermostat located in the reheated air chamber, one point of which controls the first section, while the other point controls the second section. There will be a thermostat installed in each duct leading from a fan to control the mixing dampers; this thermostat will be set to maintain, during the heating season, a constant temperature of the air as required by the room served.

On the Job

. . . a department devoted to short, practical articles on the installation, operation and maintenance of air conditioning, heating and piping equipment . . . . . .

Care Needed in Valve Operation

By. W. H. Wilson

FIG. 1—A VALVE STEM BROKEN BY EXCESSIVE PRESSURE ON THE HAND WHEEL

CARE should be exercised in operating valves, especially those that have not been opened or closed for long periods of time. Proceed slowly with a valve that does not close tight, or if closed, does not open readily. When the seating faces of the valve disc and seat are worn and cut, excessive pressure on the stem will not make the valve close tight. First of all make sure whether the valve is in the open or the closed position; stems have often been broken by mistakes being made as to whether the valve was open or closed.

Valve Obstructed; Stem Broken

Fig. 1 shows a valve stem of a good grade of rolled bronze, of sufficient strength for all practical operating purposes, that was twisted and eventually broken by an excessive amount of pressure on the hand wheel when trying to force the valve to close tight. Later it was discovered that a stud bolt from a pump valve was lodged in the recess below the seating faces of the gate, preventing the gate from taking its proper position.

The best guide in handling valves that are hard to open or close is experience. Being familiar with the problems of valve operation, much can be accomplished by the "feel" of the valve disc or gate as it meets the seat or stops in the wide open position. Time, patience and judgment are better than trying to make the valve seat tight by using an excessive amount of pressure on the stem.

How to Open a Stuck Valve

Gate valves that have been closed for some time (such as division valves) and cannot be opened readily, may be started by drilling and tapping a small hole in the lower part of the valve body, between the seating faces of the gate, and forcing kerosene or penetrating oil in the interior space, so it will come in contact with the seating faces and threads on the stem. After applying the oil the hole is plugged. Tapping this opening in the valve bonnet may be the best way to get the oil to the threads on the valve stem, as for an example, when the valve is installed with the stem in a horizontal position.

Valves located on a circuit or looped main should be checked carefully when looking for trouble in the flow through the lines. One division valve, improperly handled or out of order, may cause trouble. A large manufacturing concern had trouble in maintaining their production schedule because of a shortage of water supplied from an outside source. Without properly checking the quantity of water that should have been delivered through the underground loop water main, arrangements were made to install a larger size water meter. By the time this meter was installed the plant was working at a reduced capacity and using less water. The water pressure naturally increased and no complaints were received as to water shortage. The trouble, apparently, had been located. At a later date is was discovered that the interior gate of an 8-in. underground water valve, on one of the supply branches of the looped main leading to the water meters, had become detached from the valve stem and was closing the valve port. The plant water supply was being supplied from a smaller size section of the looped water main, with a consequent reduction in pressure and volume.

Proper checking of the water flow, regarding the size of the water mains and capacity of the meters would have directed the attention to valve trouble.

Check Refrigerating Coils

By Robert S. Wheaton*

LACK of knowledge of the properties of ammonia has, at times, resulted in the building of ammonia coils used for cooling brine which were not only inefficient but which were actually warmer in certain parts of the coils than the brine surrounding the coils. For instance, the rule-of-thumb measurement of 150 ft. of 1¼ in. pipe per ton is far from true if the pipe is not

*The Vilter Manufacturing Company, Milwaukee, Wis.

made up correctly, which is the case with long tortuous coils. A difference in pressure of 12 lb. between the inlet pressure and the outlet pressure in a coil may result in serious temperature differences. Fortunately, this fact is pretty well known, but in plants built some few years ago cases of this type are sometimes found.

Consider one of these cases: A brine tank is desired to be held around 15 F; if the coils are operated at 5 or 10 F lower than the brine temperature a fair rate of heat flow will be set up. Ammonia at 5 F has a gage pressure of 19.6 lb. If the resistance to flow of the ammonia through the coil is 12 lb., then the inlet pressure is 19.6 + 12 or 31.6 lb., and the corresponding

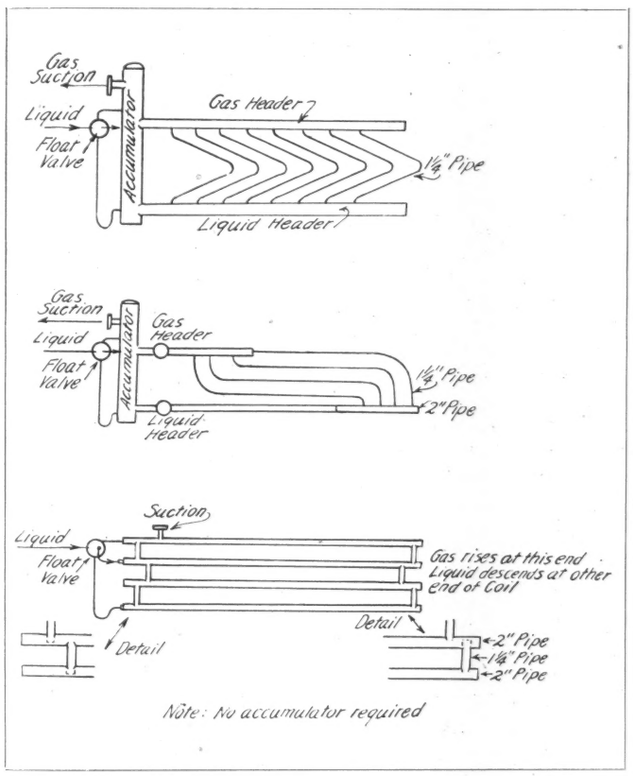


Fig. 2—Several Types of Refrigerating Coils for Brine or Ice-Making Tanks Designed for Prompt Removal of Flash Gas and Evaporated Ammonia Gas and to Promote a Natural Circulation of Liquid Ammonia to Give a High Rate of Heat Transfer

ammonia temperature is approximately 18 F, or 3 F warmer than the brine, so part of the cooling coils are actually serving to increase the temperature of the brine.

It may be thought that this is a made up case, but it actually occurred and was discovered when a thermal control feed valve was installed to regulate the feed to the coils. The valve would not feed until superheat appeared at the outlet of the coil and of course the suction gas was excessively superheated. As the owner did not wish to change the coil construction the automatic control problem was solved by using an automatic expansion valve of the pressure-reduction type with a magnetic liquid valve to shut off the liquid supply when the compressor was shut down.

A Method of Conditioning Paper

By John A. Masek

THE paper web conditioner installed in the pressroom of a Chicago newspaper consists of a brass cylinder fitted with high-velocity jet nozzles that vaporize or atomize the water with a flow of compressed air, as shown in Fig. 3. The number of jets employed depends on the sheet width of the paper. For instance, a 36-in. width might require 3 to 4 atomizing jets, and a 72-in. sheet, 6 to 8 jets. The exact number of jets depends upon the kind of paper used.

Each jet will vaporize about ½ gal. of water per hr. and require about 3 cu. ft. of compressed air per min. at 30-lb. pressure. The spray nozzles are adjustable so they may be placed to impinge the moisture directly into the surface of the paper or at an angle to the sheet. Various grades of paper require different settings. Density of the vapor can be adjusted from dry air to a heavy fog.

The vapor spray operation is synchronized with the press to prevent one spot from becoming too wet, which would break the web. Each of the heads are connected

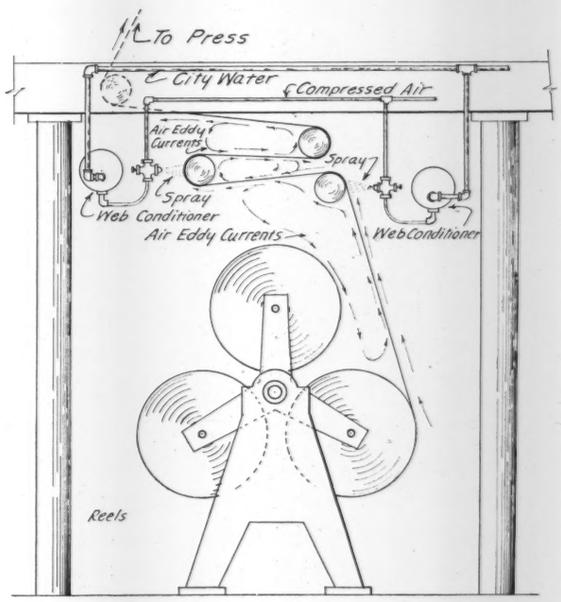
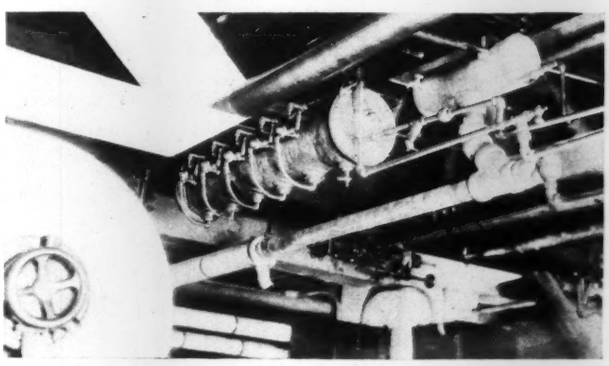


Fig. 3—The Diagram Above Shows the Method of Installation and Piping for a Paper Web Conditioner. The Installation Described Is Pictured Below



with ¾-in. piping which supplies city water and compressed air.

This method of conditioning paper was installed to insure the paper receiving the right amount of moisture despite the eddy currents set up by the speed of the paper (1,000-2,000 f.p.m.).

Advantages Gained

Since trouble from static electricity has been eliminated by conditioning the paper and the paper has been made less brittle, web breaks have been cut down.

An economy can be effected by using thinner paper because of the added strength given it. As moisture opens up the fibers of the paper it absorbs the ink more readily. It causes the ink to spread economically and a clear impression is made, with no ink pulling.

With static eliminated the sheets do not stick and the size of the conditioned paper remains the same through the press, making excellent color register possible.

Along with general room humidification this method has practically stopped ink flying, reducing slippage hazard between the presses. Paper flying is held down also, which avoids clogging the press machine parts. The elimination of both is a hygienic measure as it prevents the contamination of the air. This is well evidenced by the press room walls being kept clean over a period of time. Press blanket life is also prolonged.

General room conditioning provides the proper atmosphere for storing the rolls of paper and promotes comfortable working conditions.

Economies with Vacuum Pumps

By George T. Condron*

THE principal economies resulting from the installation of a vacuum pump in the return of a steam-heating system will fall into one of the following cases:

1 Economies resulting from a reduction of the waste due to overheating:
  A. Because of improved circulation in the system.
  B. Because of a reduction of the radiator temperatures in mild weather.

2 Economies resulting from a recovery of available heat in the condensate:
  A. Returning the hot condensate to the boiler.
  B. Using the hot condensate for domestic hot water.

Since the heat which must be supplied to a building in order to maintain any given inside temperature is practically proportional to the difference between the inside and outside temperatures, it is obvious that overheating, which increases this difference between temperatures, is a source of waste. In Case *1A* the probability of overheating is reduced because of a reduction in the probability that some of the radiators will not be carrying their share of the load. These radiators may be slacking because they are air bound or because the steam pressure is not sufficient to keep them properly supplied. A vacuum pump will suck out the air and thus put the air-bound radiator back in service. The suction of a vacuum pump will increase the available pressure difference throughout the system and thus bring back into full service any radiators which were not properly supplied before. With all of the radiators carrying their full share of the load, uniform heating of the entire building is promoted.

To evaluate a possible saving, assume one-quarter of the heating load is in a portion of a plant that must be heated to a temperature of 80 F (instead of 70 F) in order to keep the rest of the plant comfortable; further assume that the average outside temperature throughout the heating season is 40 F. Then the average difference between the temperature inside of the plant and that out-of-doors is 40 F instead of 30 F. Reducing this average temperature differential 10 degrees would result in a practically proportional decrease in the heating expense for this part of the plant, or 25%. Since this saving applies to one-quarter of the heating load, the fuel saving would be 6¼% for this case (25% times ¼ equals 6¼%).

In Case *1B* the reduction of radiator temperatures is possible because pressure at the radiators can be reduced through the operation of the vacuum pump. Through the control of radiator temperatures which this makes possible, the rate at which heat is supplied to the building (which is practically proportional to the difference between the radiator and room temperatures) can be varied to suit the need. Thus the probability of overheating during mild weather is reduced. The saving in this case could be evaluated as in Case *1A*.

Case *2* would apply only to those systems in which the hot condensate from the radiators was wasted in the sewer. It will usually be possible for the vacuum pump to transmit the condensate to some point where its heat content will be useful. A common arrangement is to return the condensate to the boiler (Case *2A*). In those cases where returning the hot condensate to the boiler is not practical, as where buildings are supplied with steam from a district heating system or other remote plant, it is possible to pump this hot condensate into the hot water part of the plumbing system (Case *2B*). The saving should be about the same as in Case *2A*, but the amount of equipment required will be greater. An installation where the saving under Case *2B* was realized is described in the September, 1930, HEATING, PIPING AND AIR CONDITIONING in the article "Condensate from Heating System Used as Hot Water Supply for Building."

*Mechanical engineer, Condron & Post, Chicago, Ill.

"Refrigerating Data Book" Published

The first edition of the "Refrigerating Data Book" has just been published by *The American Society of Refrigerating Engineers,* 37 W. 39th St., New York City. It contains 435 pages of technical data and information, including numerous diagrams, on fundamentals, principles, refrigerating machinery, insulation and containers, refrigerating equipment, applications (including air conditioning), safety code, conversion tables, glossary, etc. In addition there is a catalog section of approximately 100 pages and a directory of *A. S. R. E.* members.

The book is attractively and substantially bound; its cost in the United States is $3.50. According to the preface, it is planned to be the first of a series of biennial engineering handbooks.

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

1895

JULY 1932

Material in this Section prepared by Publication Committee: F. D. Mensing, *Chairman;* E. O. Eastwood and F. E. Giesecke A. V. Hutchinson, Manager of Publications

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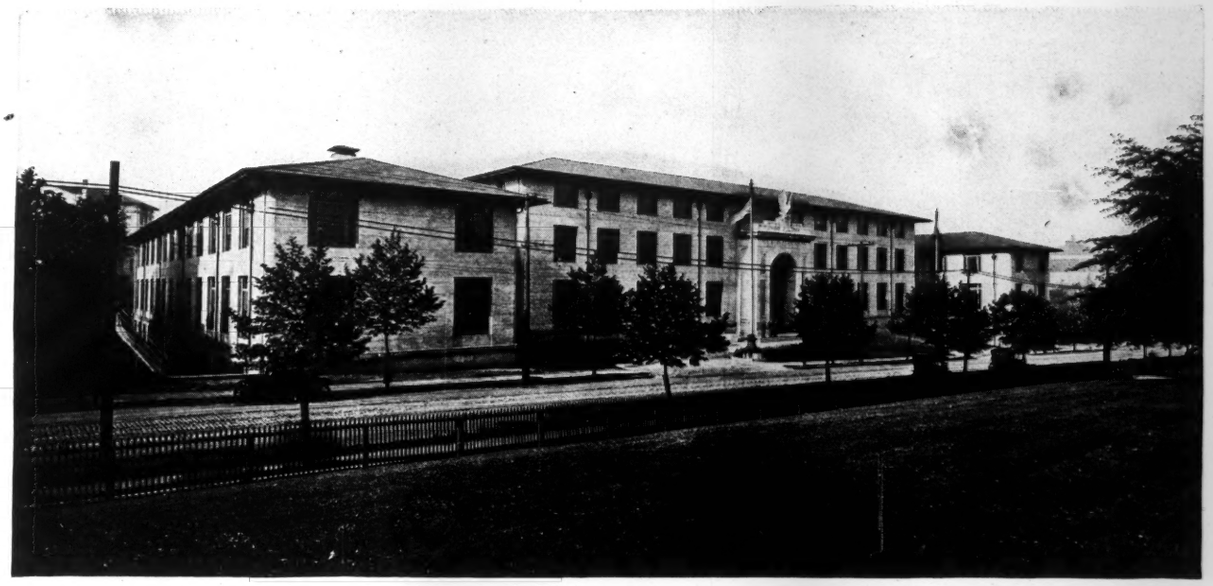

PITTSBURGH EXPERIMENT STATION OF THE U. S. BUREAU OF MINES WHERE THE RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS IS LOCATED

Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage

By F. C. Houghten[1] (*MEMBER*) and Paul McDermott[2] (*NON-MEMBER*), Pittsburgh, Pa.

THE development of the Code for Heating and Ventilating Garages[3], adopted by the Society in 1929, made apparent a need for additional facts and information concerning the life and fire hazards in garages resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors found in garages, and means for satisfactorily and economically eliminating such hazards. A Technical Advisory Committee of the Committee on Research of the Society was organized in 1928 to study the needs for research in this field.

The Technical Advisory Committee on Ventilation of Garages and Bus Terminals, under the chairmanship of E. K. Campbell, has been actively engaged in planning this work. A brief study was made at Washington University, St. Louis, Mo., in cooperation with the A. S. H. V. E. Research Laboratory, the results of which were reported in a paper entitled, Carbon Monoxide Concentration in Garages[4], by A. S. Langsdorf and R. R. Tucker. This study was made in a large commercial garage equipped with means for mechanical ventilation, but usually operated with natural ventilation only. The lack of success experienced by the investigators in controlling ventilating conditions, including the opening and closing of doors and windows, emphasized the need for additional studies in garages where the investigators are free to control the means, method and volume of ventilation, and where they are otherwise free to control the entire surroundings in accordance with the requirements of the study.

In the fall of 1931, the A. S. H. V. E. Research Laboratory was authorized to proceed with a study in Pittsburgh. For this purpose, the small single room, window ventilated, Sheraden Garage located in Sheraden, a suburb of Pittsburgh, and also the five story, underground ramp garage in the basement of the Grant Building were made available under conditions highly satisfactory and helpful to the investigation. Studies were made in these garages early in 1932. The results of the study in the Sheraden Garage are the basis of this paper, while those from the Grant Building Garage are published in a separate paper.

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[3] See A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, p. 355.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Milwaukee, Wis., June 1932.

[4] See A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930, p. 511.

Sheraden Garage

The Sheraden Garage (Figs. 1 and 2) is a two-story building with the upper floor at the street level in front, and the lower or basement floor slightly above ground in the rear. The upper floor is a solid concrete slab with no openings between the basement and the upper level, excepting through the door, *A*, leading to a stairway to the lower floor. Hence, the second floor of the garage is entirely independent of the basement floor as far as ventilation is concerned. The study was made in the upper floor of this building.

The upper floor of the garage consists of one large room or garage space, 60½ ft wide and 52 ft long. The distance from the floor to the concrete ceiling slab is 11 ft 8 in. Reinforced concrete beams, *B* and *C*, extend down from the ceiling slab 27 in. and 20 in., respectively. The entrance to the floor is by an overhead door, *D*, 8 ft wide and 9 ft high, leading into the garage space by an alley or hallway, on one side of which is an automobile showroom, and on the other side, a small storeroom. Both the first and second floors of the garage are heated by unit heaters, which were not in operation during the test.

There is no provision for mechanical ventilation, but three sides of the room are lined with horizontally-pivoted windows which open as shown by *E*, Fig. 2. Mechanical ventilation was provided for the study by arranging six 15-in. propeller fans into respective boxes which were fitted into openings made by removing the upper panes of glass from six windows. Three of these boxes, *F*, *G*, and *H*, were distributed along the right hand wall, and three, *I*, *J*, and *K*, along the left hand wall. These boxes were so arranged that the fans could be used for either supplying or exhausting air. The boxes were so designed and installed that the air was supplied or exhausted through the top of the box. Hence, the air was supplied to or taken from the garage at a point within 1 ft of the ceiling, as shown in the elevation view by fans *I* and *J*.

For the purpose of admitting air when the fans were operating as exhaust, or as a vent when the fans were operating as supply, the box, *L*, was built over window, *E*. The only opening from the room into this box was at the bottom about 1 ft from the floor, so that air was either admitted to or exhausted from the garage at the floor line as indicated. Air was also admitted to or vented from the building at the front by partially opening the overhead door, *D*, and sealing off the upper crack resulting from the partially opened door.

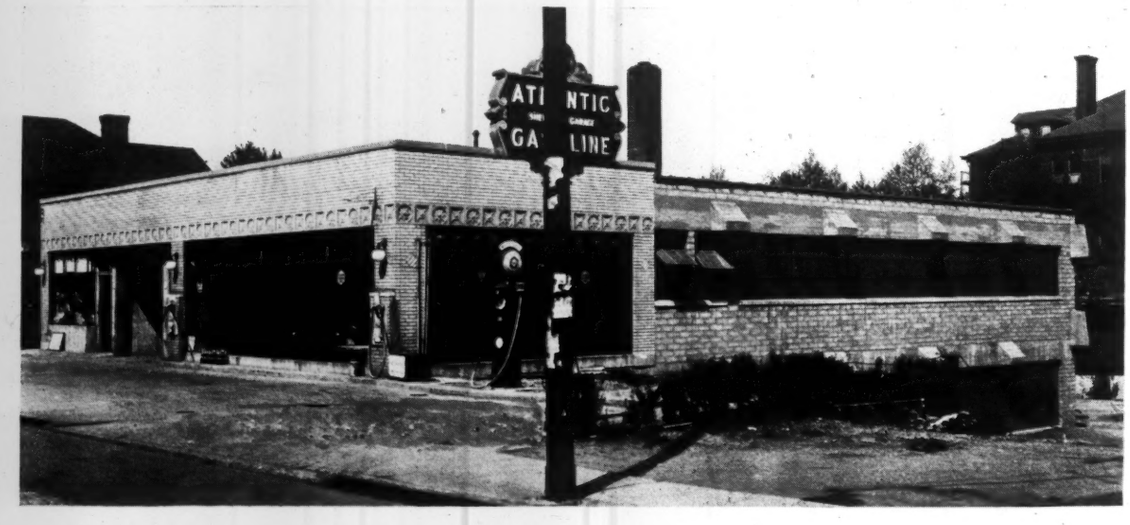
This arrangement admitted air at the front and rear at the floor line, and allowed it to escape by gravity through open windows or by forced exhaust through the fans. It also allowed for forced air supply at the ceiling by the fans, in which case the air left the garage at the front and rear floor line. Cross ventilation could also be provided by making the fans on one side supply air to, and those on the other exhaust air from, the building. When exhausting air from the building, each fan installed in its box handled 1,100 cfm of air. When supplying air to the building, each fan installed in the boxes delivered 800 cfm of air.

The temperature in the garage was approximately 60 F at the time of beginning the surveys on both nights. During the surveys, while the cars were idling, the temperature rose at times as much as 5 deg. The outdoor temperature on both nights was approximately 55 F. The first night during which the study was made (April 16, 1932), there was a wind movement of about 5 mph from the right; *i.e.*, blowing against the side of the garage next to the fans, *F*, *G*, and *H*.

Purpose of Study

The purpose of the study was essentially to determine facts of value in the ventilation of garages. Since concentration of carbon monoxide resulting from the combustion of gasoline in cars is the controlling factor affecting the ventilation of garages, the investigation became a study of carbon monoxide concentrations. In this connection, it was desirable to determine the quantity of air necessary to keep the concentration of carbon monoxide within safe limits, depending upon the garage characteristics and car activity. These facts should be of value in estimating the required quantity of air to be

FIG. 1—SHERADEN GARAGE, SHERADEN, PENNSYLVANIA



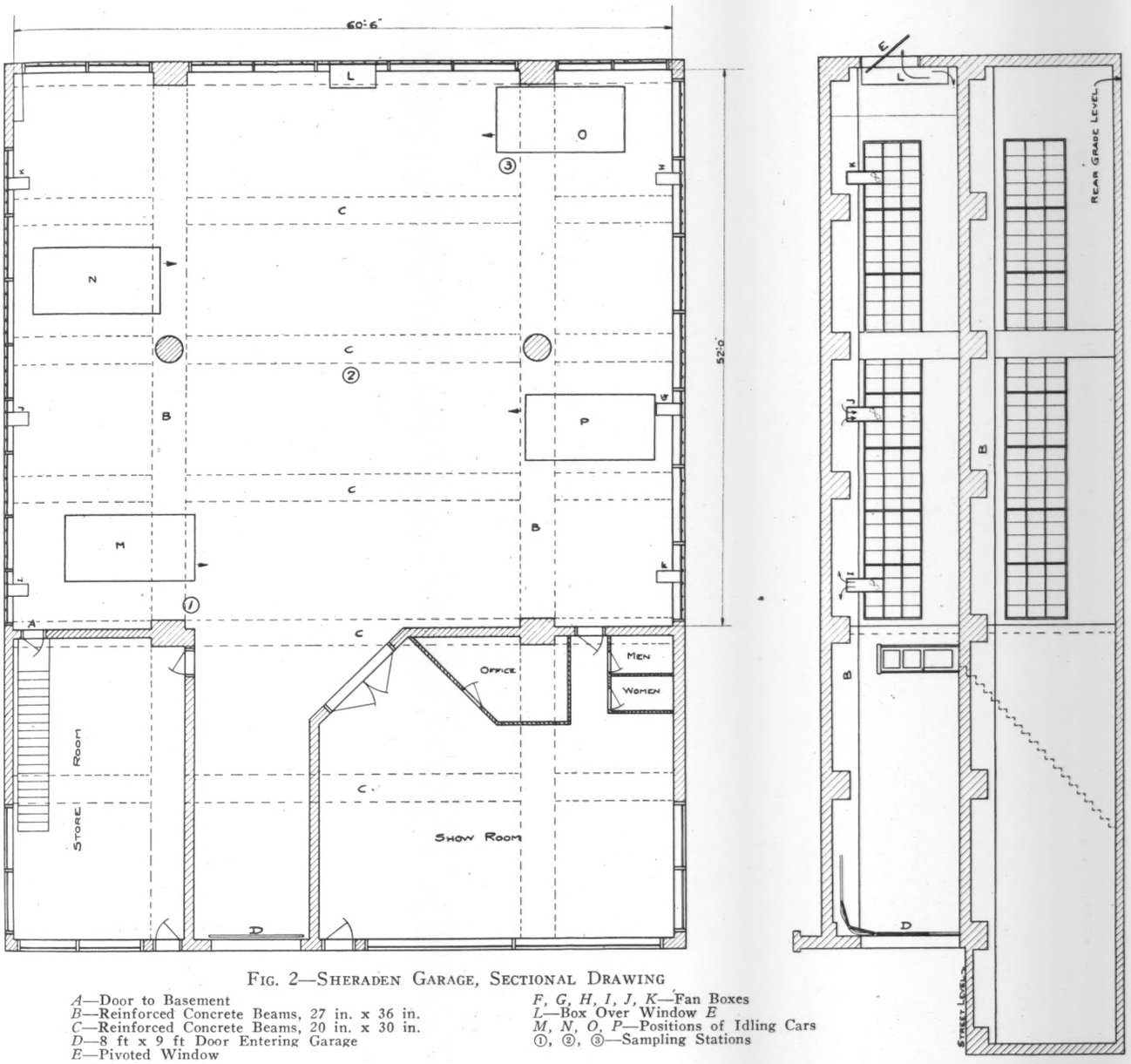


FIG. 2—SHERADEN GARAGE, SECTIONAL DRAWING

A—Door to Basement
B—Reinforced Concrete Beams, 27 in. x 36 in.
C—Reinforced Concrete Beams, 20 in. x 30 in.
D—8 ft x 9 ft Door Entering Garage
E—Pivoted Window
F, G, H, I, J, K—Fan Boxes
L—Box Over Window *E*
M, N, O, P—Positions of Idling Cars
①, ②, ③—Sampling Stations

handled in any garage ventilation problem, and in arriving at the best point for supplying and exhausting air.

Method of Attack

The study resolved itself into a series of surveys designed to give the concentration of carbon monoxide within the garage, depending upon the type and amount of ventilation and the car operation. The most convenient method of making such surveys, provided the concentration of carbon monoxide is greater than one part per 10,000, and provided the variation in concentration sought for is not less than one part in 10,000, is the *pyrotannic acid method,* developed by the U. S. Bureau of Mines.

This method of determining concentrations of carbon monoxide requires the collection of 250 cc samples of air. The standard practice is to use an ordinary rubber bulb aspirator and pump the sample into a 250 cc rubber stoppered bottle. Fifty squeezes of a 75 cc rubber bulb has been proved sufficient to give a representative sample. The samples contained in the 250 cc bottles are then taken to a laboratory for the analysis, which is completely described in a U. S. Bureau of Mines publication[5]. Briefly, the analysis consists of absorbing the carbon monoxide contained in a sample into 0.1 cc of human or ox blood diluted to 2 cc with distilled water, and comparing the resulting color, after chemical treatment with pyrotannic acid, with a standard set of colors. With proper equipment, a trained observer can analyze 25 air samples per hour. The pyrotannic acid method of analysis has a distinct advantage over other methods in that a large number of samples of air can be taken over a short period of time, representing conditions at many points through the space surveyed.

Carbon monoxide as a hazard in respired air has been studied extensively by the U. S. Bureau of Mines, the

[5] The Pyrotannic Acid Method for the Quantitative Determination of Carbon Monoxide in Blood and Air, by R. R. Sayers, W. P. Yant and G. W. Jones. Report of Investigations, Serial No. 2486, Bureau of Mines, June 1923. 6 pp.; Technical Paper 373, Bureau of Mines, 1925, 18 pp.

U. S. Public Health Service and many other organizations[6]. While the investigators have been reluctant to recommend any concentration as safe, it is generally recognized that one part of carbon monoxide in 10,000 parts of air can be breathed indefinitely without noticeable injury. Slightly higher concentrations, possibly as high as two parts in 10,000, may be endured without noticeable injury for shorter periods of time. In many of these studies carbon monoxide concentrations have been recorded as parts in 10,000, which terminology has been fairly well standardized and will be used in this paper. Hence, when any numerical concentrations are spoken of, they will be understood to mean parts in 10,000.

Surveys in the Garage

Eight surveys were made in the Sheraden Garage. In all surveys, samples of air were taken for analysis by the pyrotannic acid method at points, 1 ft, 5 ft, 7 ft 8 in., and 10 ft 8 in., above the floor at Stations *1, 2,* and *3,* Fig. 2. In all of the surveys conditions of car operation, opening and closing of windows and doors, and ventilation were under complete control of the investigators. Cars in some of the positions, *M, N, O,* and *P,* Fig. 2, were allowed to idle. All spaces along the right and left hand walls were lined with cars parked similar to idling cars; there were, however, no cars in the middle space between beams, *B*. The conditions and results of the surveys, together with any conclusions to be drawn from the findings are listed below.

Survey 29, No Ventilation, April 16, 1932, 10:00 to 10:04 p.m.

All windows and doors were closed and no mechanical ventilation was provided for this survey. Four cars, *M, N, O,* and *P,* Fig. 2, were started idling at 9:50 p.m., and samples were collected between 10:00 and 10:04. The analyses of the samples are tabulated in Table 1. The average concentration of carbon monoxide found at the four elevations are plotted in Fig. 3, giving the curve labeled *no ventilation.*

An unsatisfactory condition existed at the time of this survey, one sample indicating 15 parts of carbon monoxide per 10,000 parts of air, which concentration is extremely dangerous, except for very short exposures. In fact, the condition was such as to make physiological collapse possible in a 30-min exposure.

The visible smoke in the exhaust from the cars rose very noticeably and tended to stratify at the ceiling. However, by the time the cars had been in operation 4 or 5 min the stratification of visible smoke had lowered, so as to fill the entire garage, with little noticeable demarcation. Conditions in the garage at the time of the survey were visibly bad, and the smoke and other products of combustion in the exhaust gases irritated the throat and eyes.

Survey 30, Gravity Ventilation Through Fans, April 16, 1932, 10:28 to 10:31 p.m.

For this survey the conditions in Survey 29 were altered by reducing the number of idling cars from four to two (cars *N* and *P* were shut off), uncovering the top opening of the fan boxes so that air could pass out through the fans by gravity, raising the front door 20 in. above the floor and opening the window, *E,* in box *L*. These changes were made at 10:05. The same number of samples were taken at the same locations as in the last survey between 10:28 and 10:31. The analyses of these samples are tabulated in Table 1.

The curve, *gravity ventilation through fans,* Fig. 3, shows the average concentration at the three stations for the indicated distances above the floor. In spite of the fact that the number of cars idling was reduced by one-half, and that some ventilation was supplied, the carbon monoxide concentration in the garage had increased over the previous survey. It was visibly noticeable that the condition had not improved.

Survey 31, Gravity Ventilation Through Windows, April 16, 1932, 10:41 to 10:44 p.m.

In order to increase the effect of gravity ventilation over that shown in the last survey, 6 of the 24 windows

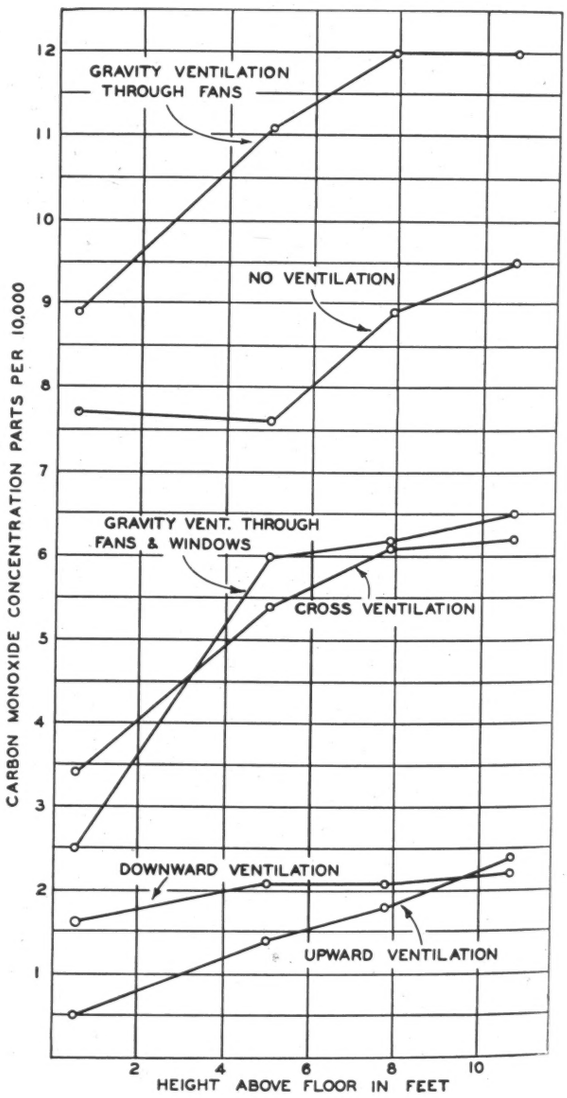


FIG. 3—VARIATION IN CARBON MONOXIDE CONCENTRATION FROM FLOOR TO CEILING IN SHERADEN GARAGE FOR DIFFERENT METHODS OF VENTILATION

[6] See Bibliography in the bulletin, Carbon Monoxide Poisoning in Industry, issued in 1930 by the Department of Labor, State of New York, pages 231 to 238.

TABLE 1—SURVEYS IN SHERADEN GARAGE

SURVEY NO.	DATE	TIME	STATION 1				STATION 2				STATION 3				AV. STA. 1, 2, 3			
			6″	5′-0″	7′-8″	10′-8″	6″	5′-0″	7′-8″	10′-8″	6″	5′-0″	7′-8″	10′-8″	6″	5′-0″	7′-8″	10′-8″
29	Apr. 16	10:00–10:06	8.0		8.0	6.7	9.4	9.4	12.0	15.0	5.7	5.7	6.7	6.7	7.7	7.6	8.9	9.5
29	Apr. 16	P. M.				7.0												
30	Apr. 16	10:28–10:31	8.0	12.0	12.0	12.0	9.4	12.0	12.0		9.4	9.4	12.0	12.0	8.9	11.1	12.0	12.0
30	Apr. 16	P. M.				12.0												
31	Apr. 16	10:41–10:44	0.3	5.7	6.7	6.7	3.8	6.7		6.7	3.3	5.7	5.7	5.7	2.5	6.0	6.2	6.4
31	Apr. 16	P. M.				6.7												
32	Apr. 16	10:57–11:02	4.7	6.7	8.0	6.7	3.8	5.7	5.7	5.7	3.3	3.8	4.7	4.7	3.9	5.4	6.1	6.2
32	Apr. 16	P. M.				9.4												
33	Apr. 22	11:35–11:40	1.9	1.9	2.5	2.5	1.9	2.1	2.5	2.5	0.8		1.3	1.9	1.7	2.0	2.1	2.4
33	Apr. 22	P. M.	2.5			2.5	2.1			3.0	0.8							
34	Apr. 22	11:48–11:52	1.3	1.9	2.5	2.5	1.9	2.5	2.5	1.3	0.8	2.5	1.3	1.9	1.6	2.2	2.1	1.9
34	Apr. 22	P. M.	1.9			2.5	1.9			1.3	1.9							
Average Survey 33, 34			1.9	1.9	2.5	2.5	1.9	2.3	2.5	2.0	1.1	2.5	1.3	1.9	1.6	2.1	2.1	2.2
35	Apr. 23	12:56–1:00	0.3	0.8	1.3	1.9	0.8	0.8	2.5	1.9		1.9	2.1	2.5	0.6	1.2	2.0	2.3
35	Apr. 23	A. M.	0.8			2.5	0.3			2.5				2.5				
36	Apr. 23	1:10–1:14	0.3	1.9	1.9	2.5	0.3	0.8	0.8	2.5		1.9	1.9	2.5	0.4	1.5	1.5	2.5
36	Apr. 23	A. M.	0.0			2.5	0.8			2.5				2.5				
Average Survey 35, 36			0.4	1.4	1.6	2.4	0.6	0.8	1.7	2.4		1.9	2.0	2.5	0.5	1.4	1.8	2.4
Average															3.4	4.6	5.1	5.4

similar to *E*, Fig. 2, were opened, 3 on the right side and 3 on the left side. The air intake in the front was enlarged by opening the overhead door to 3 ft above the floor. These changes were made between 10:31 and 10:35. The two cars, *M* and *O*, continued idling.

The same number of samples were taken at the same stations as in the last survey between 10:41 and 10:44. The results of these analyses are tabulated in Table 1, and the average for the 3 stations is plotted for the indicated distances from the floor in the *gravity ventilation through windows* curve, Fig. 3. That the gradient from the floor to ceiling falls off between 5 and 8 ft is probably due to the fact that the open windows in this case extended down to a point 5 ft above the floor.

The air condition in the garage had noticeably improved, particularly for the first 3 or 4 ft above the floor line. There was a noticeable stratification of visible smoke from 4 ft above the floor to the ceiling.

Survey 32, Cross Ventilation, April 16, 1932, 10:57 to 11:02 p.m.

Conditions for this survey were obtained by adjusting the 3 fans on the right hand side, *F*, *G*, and *H*, to blow air into, and those on the left, *I*, *J*, and *K*, to exhaust air from, the garage. The front door and all windows, including window *E*, in box, *L*, were closed, and three cars, *M*, *N*, and *O*, were idling. These changes were made immediately after the last survey, and samples were taken at 3 stations and 4 levels between 10:57 and 11:02. The analyses of these samples are tabulated in Table 1. Also, the averages for the 3 stations are plotted against distance from the floor in the *cross ventilation* curve, Fig. 3. This condition of ventilation shows a consistent gradient from floor to ceiling. The air supplied by the fans on the right side of the garage fell noticeably and rather quickly to the lower regions, and probably spread out over the floor. There was some noticeable stratification of visible smoke in the upper half of the garage, excepting for a distance of about 10 ft away from the right hand wall.

Surveys 33 and 34, Downward Ventilation, April 22, 11:35 to 11:40 p.m., 11:48 to 11:52 p.m.

For these surveys all fans were arranged to blow air

into the garage through the boxes at points one foot below the ceiling. The front door was opened 20 in. above the floor, and the window, *E*, in box, *L*, was opened; all other windows were closed. Three cars, *N*, *O*, and *P*, were started idling. However, one was found to have stopped sometime during the surveys.

These changes were made and the cars started idling at 11:10 p. m., and samples for Survey 33 were taken between 11:35 and 11:40. With no changes in conditions samples for Survey 34 were taken 12 min. later or between 11:48 and 11:52 p. m. The results of the analyses of these samples are given in Table 1, and the averages are plotted against distance from the floor in Fig. 3, giving the *downward ventilation* curve. Little gradient from the floor to the ceiling is shown. There was practically no visible smoke in the garage and no stratification at the ceiling. There appeared to be a slight stratification of smoke in some parts of the garage midway between the floor and ceiling. The visible smoke in the exhaust coming from the rear of the idling cars continued to rise toward the ceiling, in spite of the downward ventilation, although not quite as noticeably as was the case with upward ventilation.

Surveys 35 and 36, Upward Ventilation, April 23, 12:56 to 1:14 a.m.

For these two surveys the fans were all adjusted so as to exhaust air from the garage through the boxes. The front door was raised 20 in. above the floor, and the window, *E*, in box *L*, was opened; all other windows were closed. These changes were made by 12:18. Three cars, *N*, *O*, and *P*, were kept idling. Samples were taken at the four elevations above Stations *1*, *2* and *3*, between 12:56 to 1:00 a. m. Without changing conditions, a second series of samples, Survey 36, were taken between 1:10 and 1:14 a.m. Analyses of these samples are given in Table 1, and the averages are plotted in the *upward ventilation* curve, Fig. 3. There was little or no visible smoke in the garage and no noticeable stratification at any level. A marked gradient in carbon monoxide concentration is shown from the floor to the ceiling. The concentration at the breathing line was 1.3 parts in 10,000, compared with 2 parts for the same condition with downward ventilation. The concentration at the ceiling was slightly higher than it was for downward ventilation.

Discussion of Results of Surveys

Observations made of visible smoke show that exhaust gases from an automobile rise quickly upon leaving the car, due to their higher temperature. Further, high concentrations of carbon monoxide found near the ceiling indicate that high concentrations of carbon monoxide are usually found at the ceiling. These facts would seem to prove conclusively the desirability as a general rule of removing the vitiated air at the ceiling and providing the incoming air supply at the floor.

In the studies here recorded the air supplied was at a slightly lower temperature than that prevailing in the garage, which unquestionably aids in stratification of the exhaust gases at the ceiling when this cooler air is supplied at the floor line. That this same tendency of the exhaust gases to stratify at the ceiling can be relied upon if air heated above the general room temperature is supplied at or near the floor line, does not necessarily

TABLE 2—AIR REQUIRED TO DILUTE AUTOMOBILE EXHAUST GASES TO SAFE PROPORTIONS IN AIR BREATHED

(Car tested under summer conditions unless otherwise stated)

Number of Cars Tested	Type of Car[a]	Speed Miles per Hour	Average Carbon Monoxide in Exhaust Gas: Per Cent	Average Carbon Monoxide in Exhaust Gas: Cubic Feet per Hour per Car (at 65 F., 29.92 in. Hg)	Cubic Feet per Minute of Ventilating Air per Car Required to Dilute Carbon Monoxide to 1 part per 10,000 (safe for indefinite period)
			Engine	Idling	
5	5P	Car standing	7.3	39.2	6533
7[b]	7P	Car standing	6.3	35.3	5883
5[b]	1½T	Car standing	7.1	31.0	5167
4	1½–3	Car standing	2.4[c]	13.2[c]	2200[c]
3	3½–4½	Car standing	8.8	66.4	11070
6	5T	Car standing	6.8	50.5	8417
			Engine	Racing	
5	5P	Car standing	7.1	76.9	12820
7[b]	7P	Car standing	7.8	137.0	22830
5[b]	1½T	Car standing	7.7	67.9	11320
4	1½–3	Car standing	4.7	55.8	9300
3	3½–4½	Car standing	8.6	158.2	26370
6	5T	Car standing	7.5	105.2	17530
		Average Lo	ad and	Level Gr	ade
11[b]	5P	3	6.7	36.3	6050
7[b]	7P	3	6.6	53.1	8850
5[b]	1½T	3	7.8	47.4	7900
22	1½–3	6	7.3	67.6	11270
16	3½–4½	6	7.8	92.6	15430
18	5T	6	7.5	110.3	18380
9	5P	10	7.8	44.1	7350
6	7P	10	8.3	75.5	12580
7	1½T	10	7.6	58.7	9783
20	1½–3	10	7.5	104.0	17330
13	3½–4½	10	7.5	147.6	24600
12	5T	10	6.9	151.6	25270
9	5P	15	6.8	57.9	9650
6	7P	15	8.6	111.8	18630
7	1½T	15	6.7	77.2	12870
15	1½–3	15	6.2	103.9	17320
5	3½–4½	15	4.8	131.2	21870
		Full Loa	d and 3	% Grade	Upward
7[b]	7P	3	8.4	88.5	14750
5[b]	1½T	3	9.5	65.1	10850
22	1½–3	6	6.9	114.3	19050
16	3½–4½	6	6.9	148.1	24680
18	5T	6	6.1	163.3	27220
6	7P	10	8.9	117.2	19530
7	1½T	10	7.3	86.3	14380
22	1½–3	10	6.7	149.6	24930
16	3½–4½	10	6.1	88.5	14750
18	5T	10	4.9	199.3	33220
13	7P	15	8.3	146.7	24450
12	1½T	15	6.9	117.7	19620
15	1½–3	15	5.4	130.1	21680
16	3½–4½	15	4.1	181.7	30280
		Accelerating u	p 3% G	rade with	Full Load
7[b]	7P	0 to 15	6.5	163.6	27270
5[b]	1½T	0 to 15	5.4	65.9	10980

[a] Types are designated thus: 5*P* and 7*P* represent 5 and 7 passenger cars, 1½*T*, 1½-3, 3½-4½ and 5*T* represent trucks of the respective tonnages.

[b] Tested under winter conditions.

[c] These averages are undoubtedly too low to represent correctly the type of trucks. All cars were tested with carburetors set as submitted by the car owners. These four happened to be set for lean mixtures.

follow. The exhaust gases must necessarily leave the car at a temperature considerably higher than the surrounding air and will therefore rise, regardless of the temperature of the air supply admitted at the floor. The only counteracting effect which the warm air supply can have must be due to its tendency to set up convection currents which may distribute the exhaust gases throughout the garage after they have risen to the ceiling.

Any ventilating system which removes the carbon monoxide from the point where it tends to collect soon after leaving the cars will obviously result in a more economical system as regards power and heating requirements for maintaining the same average concentration of carbon monoxide throughout the building or at the breathing line. Hence, it would seem desirable, wherever possible to take advantage of upward ventilation, even though it should mean supplying the air at the floor line at a slightly lower temperature even when it must be heated.

For a condition of equilibrium between carbon monoxide production and air supply in a garage the rate at which carbon monoxide is being produced and added to the air is equal to the volume of air supplied (or exhausted) times the per cent of carbon monoxide in the garage air at the point of exhaust. This calculation can be applied to the condition for *cross ventilation* and *upward ventilation* in Fig. 3.

For *cross ventilation* 3 fans each supplied 800 cfm of air, or a total of 2,400 cfm to the garage while 3 cars were running. Assuming that 6.2 or the concentration found at the ceiling represents the concentration of carbon monoxide in the air at the exhaust openings, the amount of carbon monoxide produced by each car is equal to:

$$\frac{2400 \times 60 \times 0.00062}{3} = 29.4 \text{ cfh per car.}$$

For *upward ventilation* the 6 fans each exhausted 1,100 cfm, or a total of 6,600 cfm for the entire garage. Taking 2.4 as the concentration of carbon monoxide at the ceiling as representative of the air at the point of exhaust, the carbon monoxide produced is

$$\frac{6600 \times 60 \times 0.00024}{3} = 31.7 \text{ cfh per car.}$$

The values of 29.4 for *cross ventilation* and 31.7 for *upward ventilation* are low compared with those published by the U. S. Bureau of Mines several years ago as a result of their investigations of carbon monoxide hazards in vehicular tunnels. Table 2 is taken from the U. S. Bureau of Mines data.[7] This table indicates 37.3 cfh of carbon monoxide liberation for averages of 5 and 7 passenger cars idling, a value somewhat higher than the values computed from the results of this study. This discrepancy may be due to a difference in the condition of cars used in the two tests, a difference of car design and operation, and gasoline available between the time the U. S. Bureau of Mines survey was made about 10 years ago and the present, or it may be due to air change in the garage other than that handled by the fans.

The last column of Table 2, giving the volume of air per minute required to dilute the exhaust gases of vehicles of different types and degrees of operation to a concentration of one part of carbon monoxide in 10,000 parts of air is of value to engineers in designing the ventilating system for any garage on the basis of the cars to be handled.

The high rate at which the concentration of carbon monoxide in the air built up when cars were idling with *no ventilation* demonstrated the reason for the high prevalence of asphyxiations in single car garages. A single car idling in a one car garage, 10 ft x 15 ft x 8 ft, will raise the concentration of carbon monoxide in the air to 25.8 parts in 5 min, 51.5 parts in 10 min, and 103.0 parts in 20 min, assuming a rate of carbon monoxide production of 37 cfh per car. Collapse of a person at rest may result from concentrations of 25.8, 51.5, and 103.0 parts in 10,000 in 30, 20 and 12 min, respectively. If the person were active, collapse would occur sooner.

Acknowledgment

The authors acknowledge the kind cooperation of W. F. Pschirer of the Sheraden Garage, who made this building available for the study, and assisted in many instances to make the study a success.

[7] Ventilation of Garages, by G. W. Jones and S. H. Katz (A. S. H. V. E. Transactions, Vol. 29, 1923; pp. 341 to 346).

Code for Heating and Ventilating Garages

The A. S. H. V. E. Code for Heating and Ventilating Garages* applies to garages housing 35 or more motor vehicles with three or more walls pierced with openings; garages housing 25 or more motor vehicles, with two walls pierced with openings; garages housing four or more motor vehicles and located above ground, but having less than two walls pierced by openings and exposed to the outside; and garages housing four or more motor vehicles and located below the level of the ground.

The code states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. It specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impractical to operate such a system of natural ventilation, a mechanical system shall be used. Positive provisions shall be made to supply 1 cu ft of air per minute from out of doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage.

Where positive systems of exhausting air are used, the code states that the exhaust openings shall be not more than 24 in. above the floor, and not more than 50 ft apart. With garages having a capacity of not less than four or more than 35 cars within the scope of these regulations, air exhaust stacks may be considered as positive, provided they have not less than 15 sq ft of steam heating surface for each square foot of duct area, and not less than 1 sq ft of free area through both heating coil and duct for each 350 sq ft of floor area. Such an exhaust duct shall discharge above the roof and extend in any case to a height of not less than 15 ft above the heating coils.

*See A. S. H. V. E. Transactions, Vol. 35, 1929, p. 355.

Tests of Convectors in a Warm Wall Testing Booth

By A. P. Kratz[1] (*MEMBER*), M. K. Fahnestock[2] (*NON-MEMBER*) and E. L. Broderick[3] (*NON-MEMBER*), Urbana, Ill.

This paper is the result of research conducted at the University of Illinois in cooperation with the A. S. H. V. E. Laboratory

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the Director. This work is carried on in the Department of Mechanical Engineering under the direction of A. C. Willard, Professor of Heating and Ventilation and head of the department. This paper includes part of the results from one year's work constituting a continuation of a general research program devoted to the study of heating rooms with various types of direct steam and hot water radiators and convectors and the material will be incorporated in a bulletin of the Engineering Experiment Station.

THE A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[4] presented at the Semi-Annual Meeting of the Society, Swampscott, Mass., June 1931, provides for tests to be run in a warm wall booth of stated construction and dimensions. It further specifies that the standard temperature for the steam in the heating unit shall be 215 F and for the air at the inlet, 65 F, and that the temperature of the air at the inlet shall be not less than 60 F nor more than 80 F. For tests run with inlet air temperatures other than 65 F, but conforming with these limiting temperatures, the heat output obtained by test shall, according to this code, be reduced to a standard heat output by multiplying the actual heat output by the correction factor:

$$C = \left[\frac{150}{t_s - t_i} \right]^{1.3} \qquad (1)$$

where

150 = the temperature difference between steam at 215 F and inlet air at 65 F
t_s = the temperature of steam during the test
t_i = the inlet air temperature during the test

This correction factor is the one commonly used in the case of cast-iron radiators and its validity has been well established when used in this connection.

At the time the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) was presented, two points remained obscure to a certain extent. The validity of extending the application of the correction factor derived for cast-iron radiators to include convectors had not been definitely proved. A correlation between convector tests run in a warm wall booth and similar tests run in rooms simulating actual service conditions had not been experimentally demonstrated. The testing program outlined in this

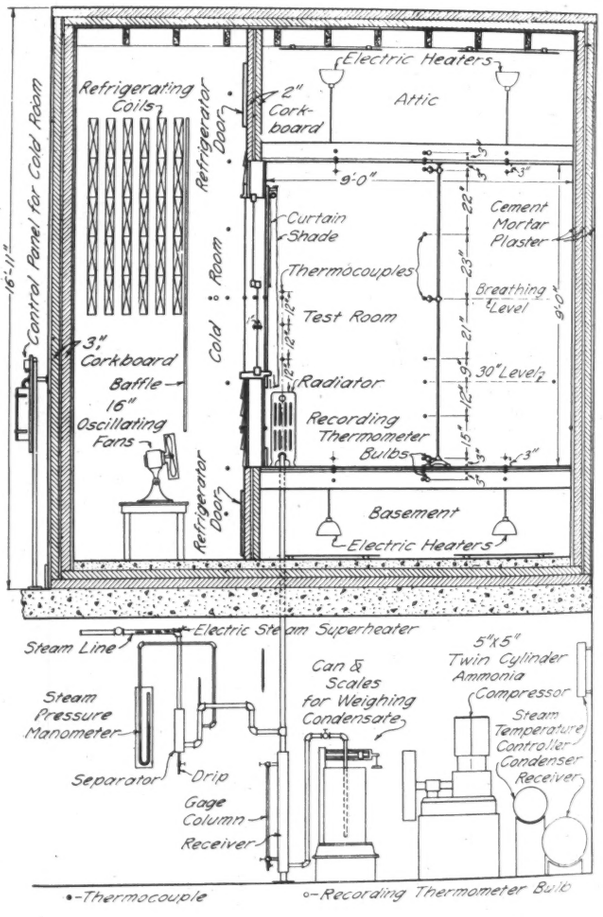


FIG. 1—ELEVATION SECTION OF LOW TEMPERATURE TESTING PLANT

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[2] Special Research Assistant Professor in Mechanical Engineering, University of Illinois.
[3] Research Assistant in Mechanical Engineering, University of Illinois.
[4] *Heating, Piping and Air Conditioning*, August, 1931, p. 695.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932.

paper was therefore undertaken in order to obtain information concerning these two points. For this purpose 225 tests were run, including tests on four distinct types of convectors and one type of cast-iron radiator, with each convector or radiator tested both in the warm wall booth and under service conditions in the low temperature testing plant, or more briefly designated as the *cold room.*

Description of Apparatus

An elevation section of the low temperature testing plant is shown in Fig. 1. This plant has been fully described in previous publications[5]. The test rooms in this plant were 9 ft by 11 ft with 9-ft ceilings.

A vertical section of the warm wall test booth showing the location of the booth with respect to the large room of the Mechanical Engineering Laboratory, in which it was erected, is shown in Fig. 2, and a detail of the piping for the heating unit and the apparatus for measuring steam condensation is shown in Fig. 3. This booth was constructed in accordance with the specifications given in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) and was 12 ft by 13 ft 4 in., with a 9-ft ceiling. For a few tests, the two-pipe steam connection shown for the warm wall booth in Fig. 3 was replaced with a one-pipe connection similar to the one shown for the low temperature testing plant in Fig. 1.

In both the test room and the test booth, the room temperature at various levels was measured on the central vertical axis of the room or booth. The temperature of the air at the inlet of the convector was measured at a distance of 3 in. in front of the inlet at at least three points spaced from 8 to 12 in. apart, midway between the top and bottom of the inlet opening. In the case of the test room in the low temperature testing plant all temperatures were measured by means of thermocouples in order to avoid any necessity for entering the rooms and thus disturbing the test conditions. In the case of the test booth, all temperatures were measured by means of mercury thermometers as specified in the code. In this case, the booth had one open side, and was surrounded by temperature conditions similar to those in the booth itself. Hence, no disturbance was caused by the observer entering the booth.

Four distinct types of convectors and one type of cast-iron steam radiator were tested. For three of the types of convectors, two different sizes of heating units were tested for each type, thus making in all 8 different units tested. The types and dimensions of all convectors are shown in the table in Fig. 4 and in the insets in Figs. 5 to 12.

[5] University of Illinois, *Engineering Experiment Station Bulletins Nos.* 192 and 223; Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929); Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock (*Heating, Piping and Air Conditioning,* July, 1931, pp. 621-625); Performance of Convector Heaters, by A. P. Kratz and M. K. Fahnestock (*Heating, Piping and Air Conditioning,* April, 1932, pp. 283 and 284).

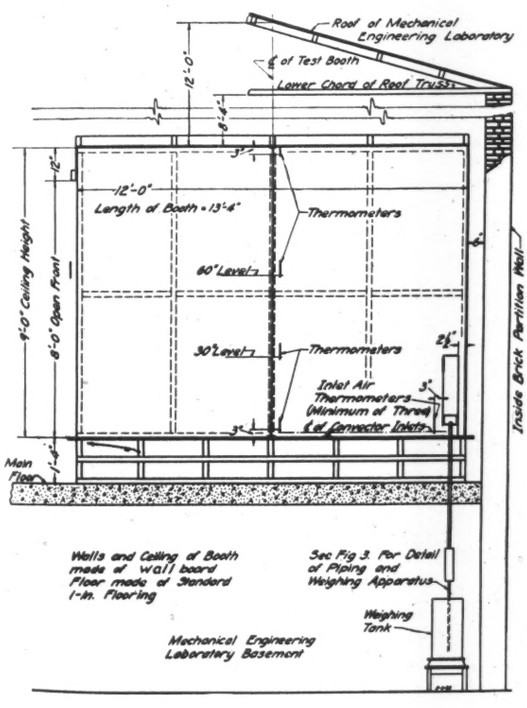

FIG. 2—SECTION OF WARM WALL TEST BOOTH

Test Procedure

Each convector was tested in both the low temperature testing plant and the warm wall test booth with the full amount of heating surface effective; that is, no part of the heating surface or air passages was either wrapped or blocked.

In the case of the low temperature testing plant, the temperature in the cold room was maintained at about —2.0 F and one of the exposed walls was subjected to an equivalent wind velocity of approximately 10 mph. The temperature above the ceiling was maintained at 62 F and the air in the basement at such a temperature that the upper surface of the floor was approximately 2 deg warmer than the lower surface. Each convector was allowed to establish whatever temperature conditions were necessary in the room in order to maintain equilibrium between the heat loss from the room and the heat output of the particular convector. In selecting the sizes of the convectors, however, the selection was limited to sizes that would not either overheat or underheat the room an unreasonable amount; that is, temperatures above 75 F or below 60 F at the 30-in. level were considered as highly undesirable.

In the case of the test booth it was considered advisable to obtain a curve for each convector, establishing the relation between the steam condensation and the temperature of the air at the inlet of the convector. Tests were therefore run at different inlet air temperatures, varying over a range of 60 F to 90 F. In order to accomplish this, the large room in which the test booth was erected was heated or cooled to a temperature approximating the desired inlet temperature, and the convector was allowed to establish the temperature conditions in the booth necessary for thermal equilibrium.

For both the test room and the test booth, no test observations were made until conditions had remained constant for several hours, as indicated by readings of all thermocouples or thermometers. When the required thermal constancy had been attained, the condensate

was weighed over a period of one hour, and no test was accepted if the condensate showed more than 2½ per cent deviation in the successive 10-min increments of weight. At the end of each test, a separate test was run to determine the condensation in the piping alone, and the total condensate was corrected by subtracting the amount so determined.

Results of Tests

The results of the tests made in the warm wall booth for all of the heating units tested are shown as full line curves in Figs. 5 to 12. These tests are represented by the unnumbered points, and the curves indicate the mean trend of these points. Additional points shown

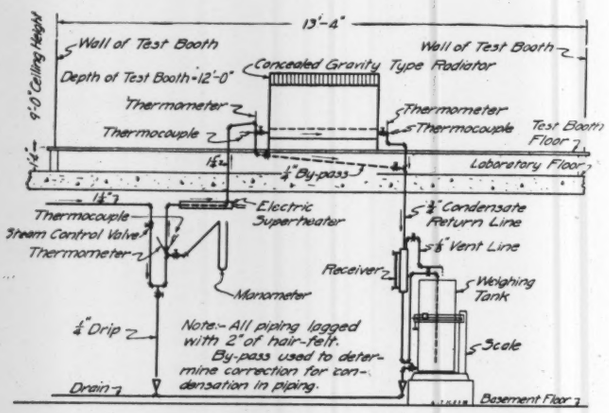

FIG. 3—DETAIL OF PIPING FOR WARM WALL TEST BOOTH

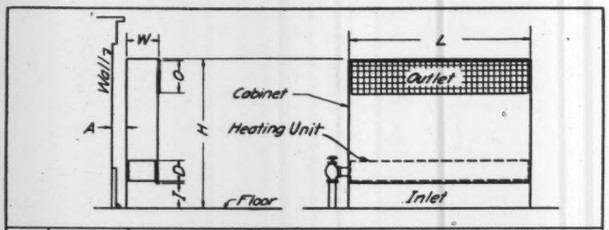

Fig. No	Convector No.	Type of Heating Unit	Dimensions in Inches A	W	H	L	I	O	D
5	8-Sec. 26"-5 Tube	Radiator	2½	8	26	20	—	—	—
6	2	Non-ferrous	2½	5	30⅜	31	6	6	6
7	3	Non-ferrous	2½	7½	30⅜	21	6	6	6
8	1	Cast Iron	2½	5½	26	37½	4	5	14⅜
9	13	Cast Iron	2½	4	26	37½	3	5	14⅜
10	6	Non-ferrous	2½	5½	28	30⅜	4	5	2
11	5	Non-ferrous	2½	5½	18	36	4	5	2
12	7	Bimetallic	2½	6⅛	28⅜	22⅜	4⅛	3⅛	20

FIG. 4—DIMENSIONS OF HEATING UNITS TESTED

near each curve, specifically designated by test numbers, represent the results from the tests made in the room of the low temperature testing plant. Duplicate tests were always made in the latter case, and when no change in conditions occurred, greater differences than 2.0 per cent in the steam condensations for the two tests were never obtained. The differences in temperature between the steam in the heating unit and the air at the inlet of the enclosure have been used as the abscissae for all curves, but since the tests were all run with a steam

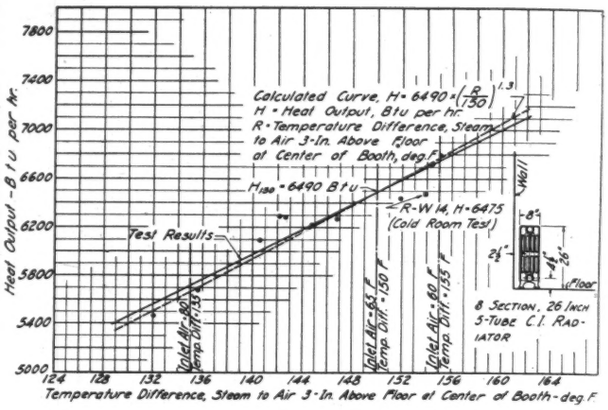
Heat Output - B t u per hr.
Calculated Curve, H = 6490 × (R/150)^1.3
H = Heat Output, B t u per hr.
R = Temperature Difference, Steam to Air 3-In. Above Floor at Center of Booth, deg. F.
H150 = 6490 B t u
R-W 14, H = 6475 (Cold Room Test)
Test Results
Wall
8"
2½"
4½"
26"
Floor
8 Section, 26 Inch 5-Tube C.I. Radiator
Inlet Air = 60 F
Temp Diff = 135 F
Inlet Air = 65 F
Temp Diff = 130 F
Inlet Air = 80 F
Temp Diff = 115 F
Temperature Difference, Steam to Air 3-In. Above Floor at Center of Booth - deg. F.

FIG. 5—PERFORMANCE CURVE FOR 8-SECTION, 26-INCH, 5-TUBE C. I. DIRECT RADIATOR

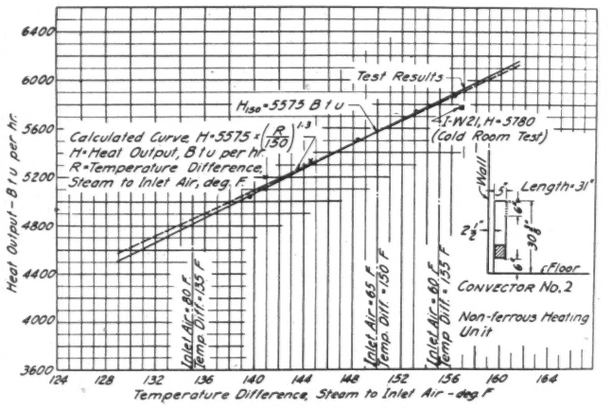
Heat Output - B t u per hr.
Test Results
H150 = 5575 B t u
A-W21, H = 5780 (Cold Room Test)
Calculated Curve, H = 5575 × (R/150)^1.3
H = Heat Output, B t u per hr.
R = Temperature Difference, Steam to Inlet Air, deg. F.
Wall
5"
Length = 31"
2½"
30½"
6"
Floor
CONVECTOR No. 2
Non-ferrous Heating Unit
Inlet Air = 60 F
Temp Diff = 135 F
Inlet Air = 65 F
Temp Diff = 130 F
Inlet Air = 80 F
Temp Diff = 115 F
Temperature Difference, Steam to Inlet Air - deg. F.

FIG. 6—PERFORMANCE CURVE FOR CONVECTOR NO. 2

temperature of 216.5 F these abscissae are also representative of the temperature of the air at the inlet, and the latter may be obtained by subtracting any temperature shown from 216.5 F.

In the case of the 8-section, 26-in., 5-tube unenclosed cast-iron radiator shown in Fig. 5, the heat output has been plotted against the difference between the temperature of the steam and the air 3 in. above the floor at the center of the booth or room. Owing to the disturbing influence of the radiator itself, temperature measurements made in the immediate vicinity of the radiator were of uncertain value, and the temperature of the air 3 in. above the floor at the center of the room was comparable with the temperature of the air at inlet observed for the convectors. In comparing Fig. 5 with Figs. 6 to 12, it is evident that the unenclosed radiator was more sensitive to stray air currents than the convectors, but it was possible to draw a curve representing a fair average trend for the points.

In order to determine the probable deviation of the corrected heat outputs, as provided for in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code), from the values that would have been obtained by tests at various temperature ranges between steam and inlet air, the calculated curves, shown as broken lines, have been superimposed on the test curves in Figs. 5 to 12. Since the

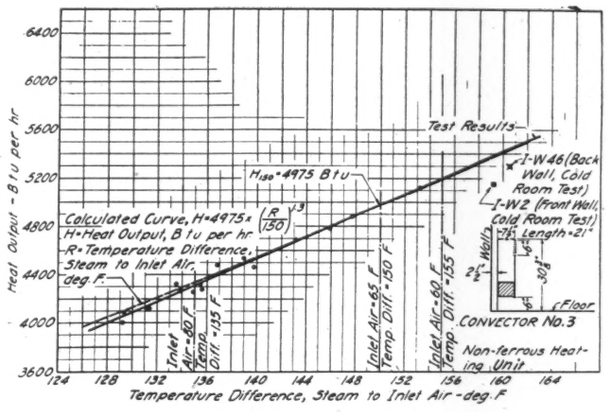
Heat Output - B t u per hr
Temperature Difference, Steam to Inlet Air - deg. F
Test Results
I-W46 (Back Wall, Cold Room Test)
I-W2 (Front Wall, Cold Room Test)
Length = 21"
Calculated Curve, H = 4975 × (R/150)^1.3
H = Heat Output, B t u per hr
R = Temperature Difference, Steam to Inlet Air, deg. F.
H150 = 4975 B t u
Inlet Air = 80 F Temp Diff = 135 F
Inlet Air = 65 F Temp Diff = 150 F
Inlet Air = 60 F Temp Diff = 155 F
Wall
Floor
CONVECTOR No. 3
Non-ferrous Heating Unit

FIG. 7—PERFORMANCE CURVE FOR CONVECTOR NO. 3

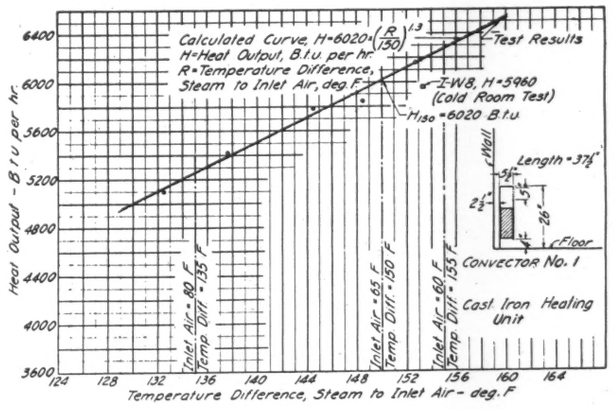
Heat Output - B t u per hr
Temperature Difference, Steam to Inlet Air - deg. F
Calculated Curve, H = 6020 × (R/150)^1.3
H = Heat Output, B t u per hr
R = Temperature Difference, Steam to Inlet Air, deg. F
Test Results
I-W8, H = 5960 (Cold Room Test)
H150 = 6020 B t u
Length = 37½"
Wall
Floor
CONVECTOR No. 1
Cast Iron Heating Unit
Inlet Air = 80 F Temp Diff = 135 F
Inlet Air = 65 F Temp Diff = 150 F
Inlet Air = 60 F Temp Diff = 155 F

FIG. 8—PERFORMANCE CURVE FOR CONVECTOR NO. 1

standard temperature range was defined as from 215 F to 65 F, or 150 deg, both the actual and the calculated curves would be coincident at this point. Hence, by passing a curve defined by the equation

$$ {}_cH = {}_aH_{150} \left[\frac{t_s - t_i}{150} \right]^{1.3} \qquad (2) $$

through the point on the test curve corresponding to the temperature difference of 150 deg, the difference between the curves at any given temperature difference would represent the approximate error in the application of the correction factor to test results obtained with that temperature difference. In Equation 2,

${}_cH$ = the calculated heat output at any temperature difference between steam and inlet air, Btu per hour
${}_aH_{150}$ = the actual heat output read from the test curve at a temperature difference of 150 deg
t_s = the steam temperature for which ${}_cH$ is to be calculated
t_i = the corresponding inlet air temperature

The deviation, or per cent error, in the application of the correction factor at the two limiting values for the temperature of the inlet air, 60 F and 80 F, as specified in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code), is given for each heating unit in Table 1. From the table it may be noted that with the exception of Convector No. 5, the correction factor is applicable to all of the heating units with an accuracy within one per cent over the specified range of inlet air temperatures of from 60 F to 80 F. In most cases, particularly in that of the unenclosed 5-tube cast-iron radiator, a slight shift in the slope of the test curve would have brought the two curves into coincidence over the whole range, and the new test curve would still be a fair representation of the trend of the points. No such shift was made, however, but the test curves were allowed to remain as first drawn, uninfluenced by the trend of the calculated curves.

The maximum deviation of 4.4 per cent occurred with Convector No. 5 which was of the same type as No. 6, for which the maximum deviation was only 0.9 per cent, and hence the difference could not be attributed to type. However, this convector had the lowest overall height of any of the heating units tested. It was only 18 in. high, while the other heating units, including the unenclosed cast-iron radiator, varied from 26 in. to 30⅜ in. Hence, it is possible that the use of the correction factor may not be applicable to convectors having overall heights outside of the range of from 25 in. to 32 in., and that further work is advisable using overall heights outside of this range.

For the discussion of the correction factor, Equation 2 has been applied to the test results obtained with a temperature difference of 150 F between steam and inlet air in order to compute the heat output that would have been obtained with a temperature difference of 135 deg, corresponding to an 80-F inlet air temperature, assuming the correction factor to be valid. In the actual ap-

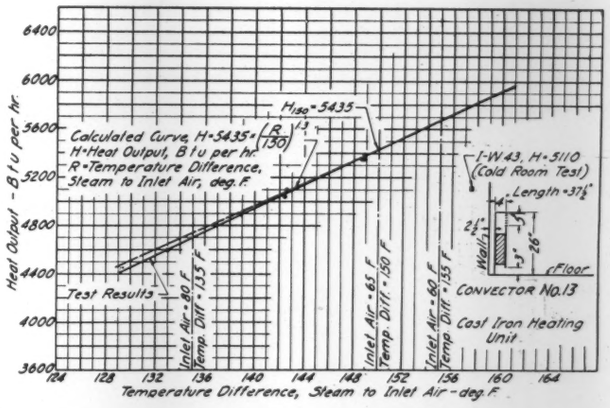
Heat Output - B t u per hr
Temperature Difference, Steam to Inlet Air - deg. F
H150 = 5435
Calculated Curve, H = 5435 × (R/150)^1.3
H = Heat Output, B t u per hr
R = Temperature Difference, Steam to Inlet Air, deg. F.
I-W43, H = 5110 (Cold Room Test)
Length = 37½"
Test Results
Wall
Floor
CONVECTOR No. 13
Cast Iron Heating Unit
Inlet Air = 80 F Temp Diff = 135 F
Inlet Air = 65 F Temp Diff = 150 F
Inlet Air = 60 F Temp Diff = 155 F

FIG. 9—PERFORMANCE CURVE FOR CONVECTOR NO. 13

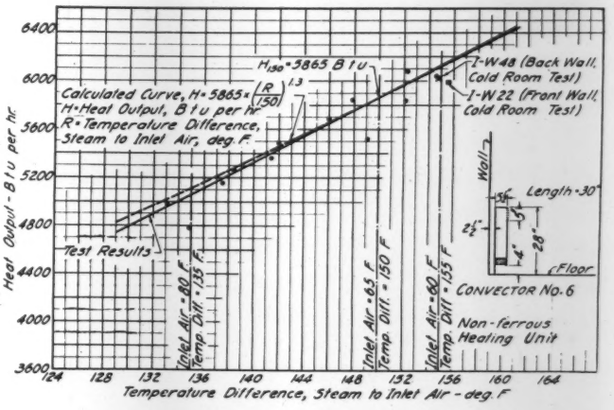
Heat Output - B t u per hr
Temperature Difference, Steam to Inlet Air - deg. F
H150 = 3865 B t u
Calculated Curve, H = 5865 × (R/150)^1.3
H = Heat Output, B t u per hr
R = Temperature Difference, Steam to Inlet Air, deg. F.
I-W48 (Back Wall, Cold Room Test)
I-W22 (Front Wall, Cold Room Test)
Length = 30"
Test Results
Wall
Floor
CONVECTOR No. 6
Non-ferrous Heating Unit
Inlet Air = 80 F Temp Diff = 135 F
Inlet Air = 65 F Temp Diff = 150 F
Inlet Air = 60 F Temp Diff = 155 F

FIG. 10—PERFORMANCE CURVE FOR CONVECTOR NO. 6

plication of the correction factor to test results the reverse process would be used. That is, the equation would be applied to the actual test results obtained with a temperature difference of 135 deg. in order to compute an hypothetical heat output that would have been obtained with the standard temperature difference of 150 deg. Since the base value of ${}_aH_{150}$ in Equation 2 would be different in the two cases, the two calculated curves would not be exactly parallel, and the difference between the actual and calculated curves at a temperature difference of 135 deg in the first case would not be identical with the difference between the actual and calculated curves at a temperature difference of 150 deg in the second case. This is illustrated in Fig. 13, from which it is evident that the difference in the two methods of treatment is purely academic since one method gives a deviation of 4.4 per cent while the other gives 4.2 per cent.

TABLE 1—DEVIATION OF CALCULATED CORRECTION FROM TEST CURVE

Fig. No.	Heating Unit No.	Type of Heating Unit	60 F Inlet				80 F Inlet			
			Heat Output Btu per Hour		100 x Ratio $\frac{{}_cH_{155}}{{}_aH_{155}}$	Per Cent Diff.	Heat Output Btu per Hour		100 x Ratio $\frac{{}_cH_{135}}{{}_aH_{135}}$	Per Cent Diff.
			By Test ${}_aH_{155}$	Calculated ${}_cH_{155}$			By Test ${}_aH_{135}$	Calculated ${}_cH_{135}$		
5	.....	5-tube C.-I. Radiator	6745	6770	100.3	+0.3	5715	5665	99.1	−0.9
6	2	Non-ferrous	5830	5810	99.7	−0.3	4820	4860	100.8	+0.8
7	3	Non-ferrous	5200	5185	99.7	−0.3	4310	4340	100.7	+0.7
8	1	Cast Iron	6270	6275	100.1	+0.1	5260	5245	99.7	−0.3
9	13	Cast Iron	5680	5670	99.8	−0.2	4700	4740	100.8	+0.8
10	6	Non-ferrous	6130	6115	99.8	−0.2	5060	5115	100.9	+0.9
11	5[a]	Non-ferrous	5575	5515	98.9	−1.1	4410	4605	104.4	+4.4
12	7	Bimetallic	6780	6805	100.4	+0.4	5740	5695	99.2	−0.8

[a] Over all height = 18 in.; overall height of all other units 26 in. to 30⅜ in.

A comparison of the heat outputs of the various units when tested first in the low temperature test plant and then tested in the warm wall booth with the same temperature for the air at inlet, may be obtained by comparing the positions of the numbered points with the positions of the curves for the booth tests in Figs. 5 to 12. For convenience, these results have been consolidated in Table 2, the last column of which shows the ratio of the heat output obtained in the low temperature testing plant to the heat output in the warm wall booth for the same temperature of air at the inlet. No very definite correlation seems to exist, but it is evident that the warm wall booth tests give heat outputs ranging from 2.4 to 12.0 per cent higher than those obtained in the low temperature testing plant under actual service conditions.

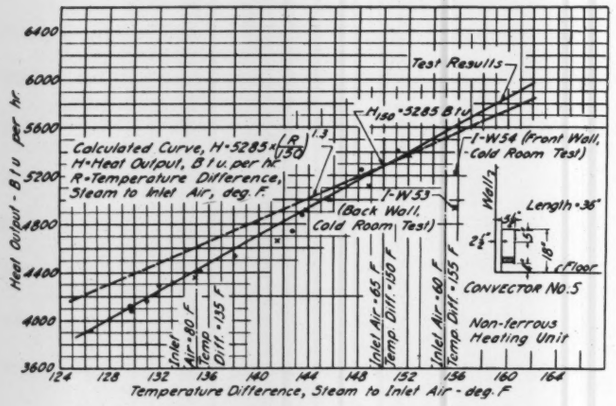
Test Results
H_{150} = 5285 B.t.u.
Calculated Curve, $H = 5285 \times (\frac{R}{150})^{1.3}$
H = Heat Output, B.t.u. per hr.
R = Temperature Difference, Steam to Inlet Air, deg. F.
I-W54 (Front Wall, Cold Room Test)
I-W53 (Back Wall, Cold Room Test)
Wall
Length = 36"
Floor
CONVECTOR No. 5
Non-ferrous Heating Unit
Inlet Air - 80° F Temp Diff - 135° F
Inlet Air - 65° F Temp Diff - 150° F
Inlet Air - 60° F Temp Diff - 155° F
Heat Output - B.t.u. per hr.
Temperature Difference, Steam to Inlet Air - deg. F.

FIG. 11—PERFORMANCE CURVE FOR CONVECTOR NO. 5

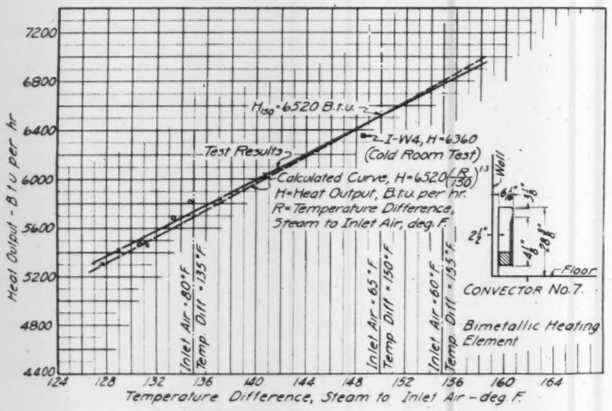
H_{150} = 6520 B.t.u.
I-W4, H = 6360 (Cold Room Test)
Test Results
Calculated Curve, $H = 6520 (\frac{R}{150})^{1.3}$
H = Heat Output, B.t.u. per hr.
R = Temperature Difference, Steam to Inlet Air, deg. F.
Wall
Floor
CONVECTOR No. 7
Bimetallic Heating Element
Inlet Air - 80° F Temp Diff - 135° F
Inlet Air - 65° F Temp Diff - 150° F
Inlet Air - 60° F Temp Diff - 155° F
Heat Output - B.t.u. per hr
Temperature Difference, Steam to Inlet Air - deg. F.

FIG. 12—PERFORMANCE CURVE FOR CONVECTOR NO. 7

Practically all of the tests in the warm wall booth were run with the two-pipe steam connection shown in Fig. 3, while in the low temperature testing plant a one-pipe connection was used. In order to determine whether the difference in heat outputs could be attributed to this difference in piping connections, the two-pipe connection in the warm wall booth was replaced with a one-pipe connection identical with the one shown in Fig. 1, and a series of tests was run on each of Convectors Nos. 2, 5 and 7. These results are shown by the crosses in Figs. 6, 11 and 12. The location of these points with respect to the other points, indicates that no difference in results could be attributed to the difference in steam connections.

During the test periods in the low temperature testing plant the rooms were closed and readings were made with thermocouples wired to a potentiometer located outside of the rooms. Hence, no disturbance in conditions could be brought about by observers entering the rooms, or by air currents in the main laboratory. In the case of the warm wall booth, the observer entered the booth to read thermometers, and since one side of the booth was open, it was possible that air currents in the main laboratory might be reflected in the booth. A desk fan was therefore placed in the booth and located in such a way that, while the air current was not directed against the convector, the disturbance of the air in the booth

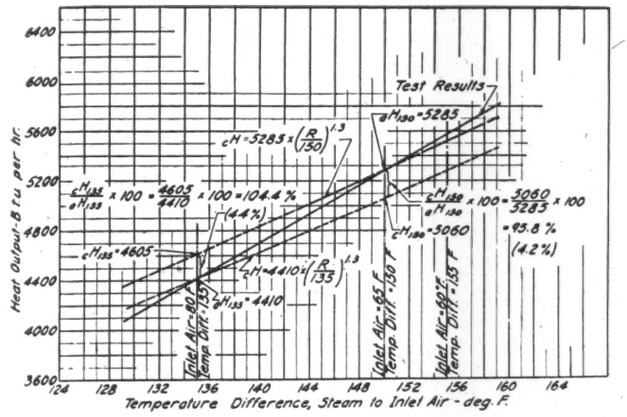

FIG. 13—SAMPLE APPLICATION OF CORRECTION FACTOR

was much greater than any that could be brought about by the observer entering the booth, or by stray air currents occurring in the large room in which the booth was located. The results from two tests indicated that the booth was comparatively free from the influence of stray air currents. Furthermore, no evidence could be obtained to indicate that the air motion in the large room was tending to circulate air into the booth at the bottom and out at the top, thus acting as a pump to increase the air circulation through the radiator. Therefore, differences in test results can hardly be attributed to these causes.

The essential difference between the influence of the rooms in the low temperature testing plant and of the warm wall booth on the performance of the heating units seems to arise from the fact that in the test room the heating unit must establish its own environment, limited by the necessity for establishing thermal equilibrium between the heat output from the heating unit and the heat loss from the room, while in the warm wall booth the heat generated can escape freely, and the environment is established more or less independent of the heat output of the heating unit. This is reflected to a certain extent by the fact that the temperature gradient from

TABLE 2—COMPARISON OF PERFORMANCE OF CONVECTORS TESTED IN WARM WALL BOOTH AND IN LOW TEMPERATURE TESTING PLANT

Fig. No.	Test No. (Cold Room)	Convector No.	Type of Heating Unit	Overall Dimensions, Inches: Width	Height	Length	Location	Btu Output per Hour	Inlet Air Temp., (Deg Fahr)	Temp., Diff. Steam to Air	30-in. Level Temp., (Deg Fahr)	60-in. Level Temp., (Deg Fahr)	$H_{Cold\ Room}$ / H_{Booth} x 100
5	*R*-W14	8-Sec. 26", 5-Tube, C. I. Rad.		8	26	20	Cold Room, Front Wall	6475	62.6[a]	153.9[b]	71.3	78.5	96.8
		"	"	8	26	20	Booth	6690	62.6[a]	153.9[b]	.....	.....	.....
6	*I*-W21	2	Non-ferrous	5	30⅜	31	Cold Room, Front Wall	5780	59.7	156.8	67.3	76.4	97.6
		2	"	5	30⅜	31	Booth	5920	59.7	156.8	.....	.....	.....
7	*I*-W2	3	Non-ferrous	7½	30⅜	21	Cold Room, Front Wall	5150	57.3	159.2	63.5	73.6	95.6
		3	"	7½	30⅜	21	Booth	5385	57.3	159.2	.....	.....	.....
	I-W46	3	"	7½	30⅜	21	Cold Room, Back Wall	5300	56.0	160.5	62.7	73.4	97.3
		3	"	7½	30⅜	21	Booth	5445	56.0	160.5	.....	.....	.....
8	*I*-W8	1	Cast Iron	5½	26	37½	Cold Room, Front Wall	5960	63.2	153.3	70.8	77.4	96.4
		1	"	5½	26	37½	Booth	6185	63.2	153.3	.....	.....	.....
9	*I*-W43	13	Cast Iron	4	26	37½	Cold Room, Front Wall	5110	58.9	157.6	65.1	72.2	88.0
		13	"	4	26	37½	Booth	5810	58.9	157.6	.....	.....	.....
10	*I*-W22	6	Non-ferrous	5½	28	30	Cold Room, Front Wall	5990	60.9	155.6	69.2	77.8	97.2
		6	"	5½	28	30	Booth	6165	60.9	155.6	.....	.....	.....
	I-W48	6	"	5½	28	30	Cold Room, Back Wall	6025	61.7	154.8	70.1	79.5	97.8
11	*I*-W54	5	Non-ferrous	5½	18	36	Cold Room, Front Wall	5230	60.7	155.8	67.8	73.5	93.1
		5	"	5½	18	36	Booth	5620	60.7	155.8	.....	.....	.....
	I-W53	5	"	5½	18	36	Cold Room, Back Wall	4940	60.7	155.8	67.8	74.8	87.9
12	*I*-W4	7	Bimetallic	6⅛	28⅜	22⅝	Cold Room, Front Wall	6360	67.9	148.6	71.2	80.1	99.0
		7	"	6⅛	28⅜	22⅝	Booth	6440	67.9	148.6	.....	.....	.....

[a] Temperature of air 3 in. above floor at center of room or booth.
[b] Temperature difference between steam and air 3 in. above floor at center of room or booth.

floor to ceiling in the room in the low temperature testing plant was much greater than that in the warm wall booth. Just what bearing this has on the performance of the heating units when the temperatures of the air at the inlets are the same in the two cases, is not apparent, but it seems probable that the heating units tested in the low temperature testing plant must all be of such a size as to produce some definite temperature at a definite level in the test room, in order for the heat output to be comparable with that obtained in the warm wall booth.

In order to determine whether such a correlation was possible, the ratio of the heat outputs obtained in the low temperature testing plant to those obtained in the warm wall booth were plotted against the corresponding temperatures at the 30-in. level obtained in the low temperature testing plant, as shown in Fig. 14. While the correlation can not be considered as very satisfactory, some marked tendencies are indicated. As the temperature at the 30-in. level in the test room became higher, the ratio of the heat outputs for tests run with the same inlet air temperature in the test room and the booth also became higher, and this ratio approached 1.0 for a temperature of approximately 69 F at the 30-in. level. The ratio for Convector No. 3 seems out of line with the others, but even taking this into consideration, it seems evident that if all of the heating units had been of such a size as to produce a temperature of 68 or 69 F at the 30-in. level in the test room of the low temperature testing plant, the heat outputs obtained in this plant would have all been within 10 per cent of those obtained in the warm wall booth with the same temperature for the air at the inlet. Hence, the evidence from these tests seems to indicate that, if a heater is rated by means of tests made in a warm wall booth according to the specifications of the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code), the heat output under service conditions in an actual room will be within 10 per cent of the rating, provided that the heating unit produces a temperature of approximately 68 F at the 30-in. level in the room.

It is possible that the heat output as determined in the warm wall booth should be multiplied by an application or service factor of approximately 0.95, as indicated by the curve in Fig. 14, in order to provide for deviations in actual service. However, the data are too limited at present to make the recommendation of such a factor advisable, and, particularly in the cases of unusual types of heating units, tests should always be required both in the warm wall test booth and in some form of low temperature testing plant subjecting the heating unit to actual service conditions, in order to establish application or service factors.

Conclusions

The following conclusions may be drawn as applying to the data obtained in these tests:

1. The correction factor, $C = \left[\frac{150}{t_s - t_1} \right]^{1.3}$ for reducing the heat output of a convector under test conditions to the heat output under standard conditions, with steam temperature of 215 F and inlet air temperature of 65 F, is applicable within a probable error of one per cent for temperatures of the air at the inlet between 60 F and 80 F, and for overall heights of from 25 in. to 32 in. with a steam temperature of approximately 215 F.

2. For overall heights outside of the range of 25 to 32 in. it is possible that the error resulting from the application of the correction factor does not exceed 5.0 per cent.

3. The heat output of a convector under service conditions in an actual room will be within 10 per cent of the heat output determined from tests in a warm wall booth, with the same temperature for the steam and for the air at the inlet in both cases, provided that the size of the convector is sufficient to heat the actual room to a temperature of approximately 68 F at the 30-in. level.

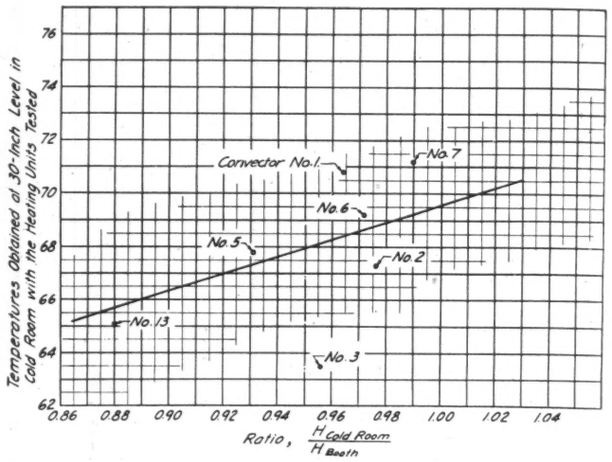

FIG. 14—CORRELATION CURVE FOR PERFORMANCE IN LOW TEMPERATURE TESTING PLANT AND WARM WALL BOOTH

Axial and Radial Flow Fans

Axial flow fans are made with various numbers of blades of a variety of forms. The blades may be of uniform thickness (sheet metal), either flat or cambered, or may be of varying thickness of so-called aerofoil section (airplane propeller type). Where a propeller fan is intended for operation at comparatively high pressures the hub is sometimes enlarged in the form of a disc and the fan is known as a *disc fan.*

Radial flow or *centrifugal fans* include steel plate fans, pressure blowers, cone fans, and the so-called multiblade fans. All the foregoing types have variations which may be obtained by modification of the proportions or change in the curvature and angularity of the blades.

The angularity of the blades determines the speed characteristic, the forward curve corresponding to slow speed and the backward curve to high speed operating characteristics.

A wide variation exists in the demands which have to be met by fan installations. A fan may be required to move large quantities of air against little or no resistance or it may be required to move small quantities against high resistances. Between these two extremes innumerable specific requirements must be met. In general, fans of all types can be made to perform the same duty, although mechanical difficulties, noise or lack of efficiency may limit the use to one or another type.

Investigation of Air Outlets in Class Room Ventilation

By G. L. Larson[1] (*MEMBER*), D. W. Nelson[2] (*MEMBER*) and R. W. Kubasta,[3] (*NON-MEMBER*), Madison, Wis.

This paper is the result of research conducted at the University of Wisconsin in cooperation with the A. S. H. V. E. Research Laboratory

THE first section of this paper which appeared in the June 1932 issue outlined the purpose of the investigation and contained a description of the buildings and ventilating systems, details of the calibration testing, results at Dudgeon School and observations at the Mechanical Engineering Building.

Tests with Motor-Blower Supply

The unit ventilator capacity is sensitive to the building up of pressure. The calibration tests in the laboratory showed this by a decrease in capacity of about 150 cfm for an increase of 0.05 in. water in pressure drop across the unit. In order to study the effect of outlet vents under different conditions of air supply, the motor-blower used for calibrating of anemometers on grilles and louvers was moved up into the room and arranged for an outside air supply. The air delivery was measured by Pitot tube traverses. Fig. 11 shows this temporary arrangement at the rear of the class room. No provision was made for heating the incoming air as was done in the case of the unit ventilator installation. In both cases the air volumes were corrected to 70 F before making comparisons.

In a typical test of the motor-blower, the air delivery to the room was 910 cfm with the outlet vent open. The amount of this measured at the door louvers was 78 per cent. Sealing of the door louvers resulted in only a three per cent reduction. Sealing of the door perimeter reduced the capacity to 89 per cent. Tests with the blower inlet partially throttled so as to obtain lower air delivery showed even less falling off in air delivery with progressive sealings. In the lowest run taken, 545 cfm were obtained with nothing sealed; with the outlet vent, door and window perimeters sealed, the capacity was reduced only to 520 cfm. Fig. 12 shows the drop in pressure from the class room to the corridor as plotted with the capacities of the various tests. This pressure drop was less than 0.01 in. water with the delivery of 1000 cfm. With the sealing of the door louvers, the same air delivery resulted in a 0.25-in. drop in pressure. Sealing of the door perimeter in addition resulted in a pressure drop of 0.48 in. for the delivery of 1000 cfm. The same air delivery with the additional sealing of the windows made a pressure drop of 1.04 in. water across the wall between the room and the corridor. The outlets from the corridor to the roof are large so that these pressure drops approach the pressure against which the fan was working. Measurements of the electrical input on this motor-blower showed little difference in current consumption for a delivery of 1000 cfm at 0.02 in. water and at 1.04 in. water. This fan is a pressure fan capable of working against 6 in. of water pressure and consequently was working at a low efficiency at 1.04 in. and still lower at 0.02 in. The power consumptions with this pressure blower do not give the correct power cost comparisons for a system designed for delivering air into tight rooms without outlet vents, compared to that found with the ordinary vented system.

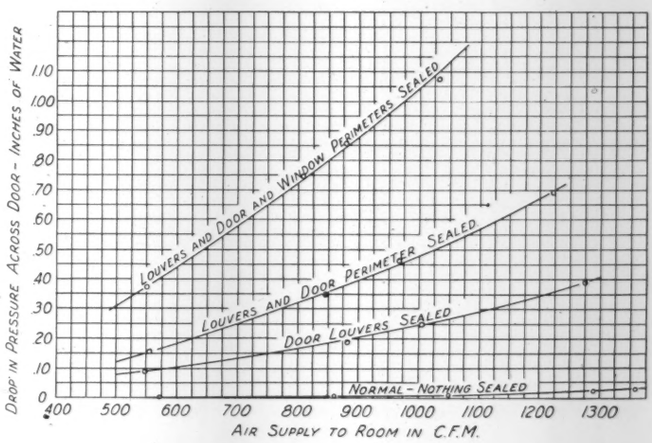

FIG. 12—RELATION OF PRESSURE DROP AND AIR SUPPLY FROM MOTOR-BLOWER FOR DIFFERENT OUTLET CONDITIONS

Central Fan System

The tests described so far were made with a unit ventilator and with a motor-blower with the inlet of the central fan system sealed. The central fan was used in most of these runs to provide normal conditions in the corridor and in the two adjoining class rooms. Because the unit ventilator was sensitive to the building up

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Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Milwaukee, Wis., June 1932.

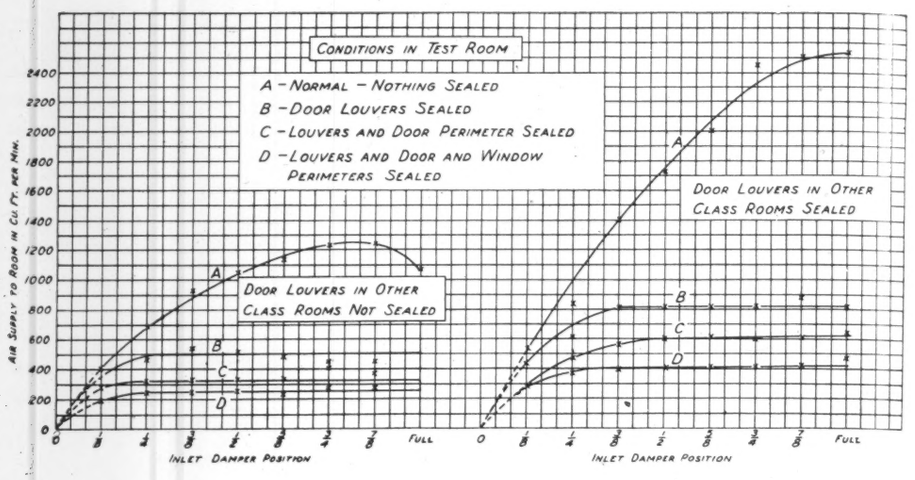

CONDITIONS IN TEST ROOM
A - NORMAL - NOTHING SEALED
B - DOOR LOUVERS SEALED
C - LOUVERS AND DOOR PERIMETER SEALED
D - LOUVERS AND DOOR AND WINDOW PERIMETERS SEALED
AIR SUPPLY TO ROOM IN CU. FT. PER MIN.
DOOR LOUVERS IN OTHER CLASS ROOMS NOT SEALED
DOOR LOUVERS IN OTHER CLASS ROOMS SEALED
INLET DAMPER POSITION
INLET DAMPER POSITION
FULL
FULL

FIG. 13—CENTRAL FAN SUPPLY UNDER DIFFERENT OUTLET CONDITIONS

of pressure, in one series the central fan was not run so as to reduce the pressure in the corridor against which the unit must discharge.

Further observations were made using the central fan for supplying the air to the class room under test. Two series of tests were made, one with the outlet louvers open in all other ventilated rooms and the other with these other louvers sealed during the period of test. Fig. 13 shows the results of these tests. The maximum capacity delivered to the room with the louvers in the other class rooms open was 1250 cfm. When these other louvers were sealed the maximum delivery to the test class room with the outlet louvers of this room open was 2530 cfm. This shows the diverting of air away from the rooms where the resistance has been increased.

In the case of the outlet vents of other class rooms not being sealed, the sealing of the door louvers in the test class room reduced the capacity from 1250 cfm to 500 cfm, which is a reduction to 40 per cent. The further sealing of the door reduced the air delivery to 26 per cent and the additional sealing of the windows reduced it to 20 per cent. These reductions are more than for the unit ventilator or motor-blower. This shows again the diverting of the air to other class rooms as resistance to out-flow is increased in the test room.

The sealing of the door louvers in the test room in the series where the door louvers of all other class rooms were sealed reduced the maximum air volume from 2530 cfm to 820 cfm, a reduction to 32 per cent. The additional sealing of the door perimeter reduced the capacity to 615 cfm, which is 24 per cent of the delivery with nothing sealed. The further sealing of the windows reduced the delivery to 16 per cent.

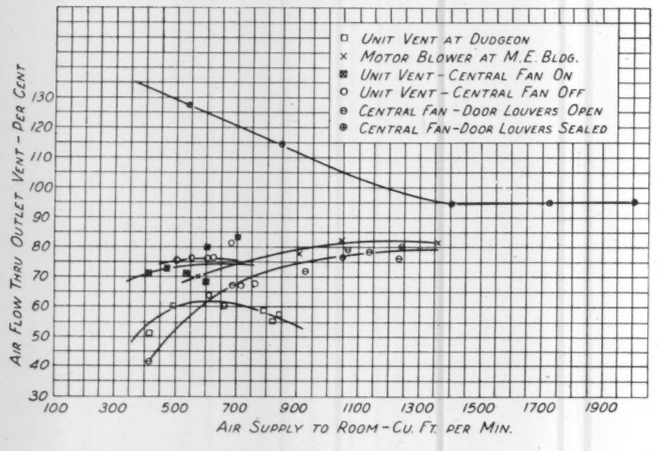

UNIT VENT AT DUDGEON
MOTOR BLOWER AT M.E. BLDG.
UNIT VENT - CENTRAL FAN ON
UNIT VENT - CENTRAL FAN OFF
CENTRAL FAN - DOOR LOUVERS OPEN
CENTRAL FAN - DOOR LOUVERS SEALED
AIR FLOW THRU OUTLET VENT - PER CENT
AIR SUPPLY TO ROOM - CU. FT. PER MIN.

FIG. 14—RELATION OF OUTFLOW THROUGH PLANNED OUTLET VENTS TO AIR SUPPLY

Readings were taken at carefully set damper positions from full open to closed by eighths. Closing of the dampers on the two runs with nothing sealed in the test room resulted in a decrease in air volume; on the various sealed runs the damper had little effect until nearly one-fourth closed.

It should be emphasized that these figures do not give the correct comparison between two ventilating systems installed in the same rooms, the two being identical except for the exits of air. In the cases cited, as the resistance was built up in the test room by the sealing of exits, the air was diverted to other class rooms. For a true comparison, the same sealing should be carried out in every ventilated room at the same time. However, the correct comparison is available for these two systems, one with and one without planned outlet vents. In Fig. 13 the maximum air delivery was 1250 cfm with the louvers open in all class rooms. This is shown in Curve *A* on the left half of the figure. In Curve *B* on the right half of the figure is shown the air delivery with the same inlet damper setting of 7/8. This delivery was 820 cfm which is 66 per cent of that obtained when outlet vents in the form of door louvers are provided. Had a lower damper setting been chosen for this comparison the per cent would have been larger as the damper reduced the unsealed delivery considerably while not affecting the sealed delivery over a wide range.

A measurement of the air delivery from all central fan inlets with all louver doors not sealed totaled 17,514 cfm. With the inlet damper wide open the test class room received 1250 cfm. This is about 7 per cent of the total air for the system. With all louvers, the outlet vents from the ventilated rooms sealed, the total capacity of the system as measured at the inlets to the various rooms was 15,130 cfm. This is 86 per cent of the delivery obtained with all door louvers open. The discharge pressure at the fan increased from 0.21 in. to 0.28 in. upon sealing the door louvers. The pressure drop on the inlet side of the fan was about 0.38 in. The

ratio of change of the total fan pressures with and without planned outlet vents then is less than that indicated by the change in delivery pressure. This accounts for the reduction being only 14 per cent for sealing of the room outlet vents.

Comparison of Air Quantities Discharged

The ability of the various equipments to deliver against the increased pressure of sealed outlets varied widely. It would be expected, however, that the proportion of air delivered to the room that left by way of the outlet louvers would be the same for all systems. That this is not true must be attributed to the inaccuracies of testing or to the influence of infiltration. With the pressures nearly balanced on the inside and outside of the building wall, the sash perimeters form one exit for air. If the pressure outside is in excess due to wind velocity, exit of air by this path is reduced, brought to zero or is replaced by a delivery of air into the room that must find its way out through other exits such as the louver door in this case.

Fig. 14 shows the air flow through the outlet vent plotted in per cent of the air supplied to the room by the different units used. Included on this sheet are also the results of the tests of the Dudgeon School class room. The outlet vent in this room is more restricted than that in the Mechanical Engineering Building class room, so it is to be expected that the curve shows a lower per cent of the air supply leaving by the planned outlet vent. Further, the small miscellaneous openings are larger and more numerous in this room.

The results shown for the central fan with the louvers open in all class rooms check fairly closely with the motor-blower results. At 1000 cfm, the air flow through the outlet vent was about 80 per cent of the air supply.

In trying to determine more accurately, the two unit ventilator curves, one with the central fan running to give normal conditions in the corridors and one with this fan not running, several check tests were made but the points scattered considerably so that there was a variation of 10 to 15 per cent at equal air supplies. The drop in pressure from the room to the corridor in these runs was less than 0.01 in. water, so that infiltration would not be checked to any extent. A wind velocity of 15 mph exerts a pressure of 0.108 in. water.

With a larger air supply as with the motor-blower and central fan, the pressure builds up, and variable infiltration should affect the points less. At the maximum delivery of the motor-blower the pressure drop from room to corridor was only 0.022 in. water. This indicates that the outlet vent was too large to allow building up enough pressure to prevent infiltration at ordinary wind velocities. With the outlet vents, the louvers in this case sealed, a delivery of 640 cfm as shown on Fig. 12 would build up a pressure sufficient to prevent infiltration with a 15-mph wind velocity. On judging the resistance to infiltration the pressure drop from room to corridor was used. The pressure drops across the outside building wall varied considerably under the influence of the wind. Since the drop in pressure from the corridor to the outside was very low because of the large outlet vents the figures used are considered satisfactory for this purpose.

The curve showing the central fan results with the louver openings in all other class rooms sealed is far removed from the other curves. It would be expected that it would lie a short distance above the other central fan curve because of the lower pressure in the corridor. That it does not is due to the influence of infiltration.

At the lowest air deliveries to the room from the fan inlet 27 per cent more air left the room by the louver opening than the fan supplied. This is 146 cfm more than the 540 cfm supplied by the ventilating system. As the air supply to the room from the fan increased the amount passing through the louvers decreased until 95 per cent of the air supply left by the louver opening. It is probable that the other 5 per cent, amounting to from 70 to 100 cfm, passed through the ½-in. space under the door. The other curves indicate that, if infiltration had not entered to a large extent into the total air entering the room, the air through the outlet vent would be about 85 per cent. The 10 per cent difference between this and the 95 per cent measured at above 1400 cfm would seem to be infiltration and would be from 140 to 200 cfm.

Two additional series of tests were made to check the measurements shown for the central fan on Fig. 12. The tests of the central fan supply with door louvers sealed in all other class rooms as shown on this figure were

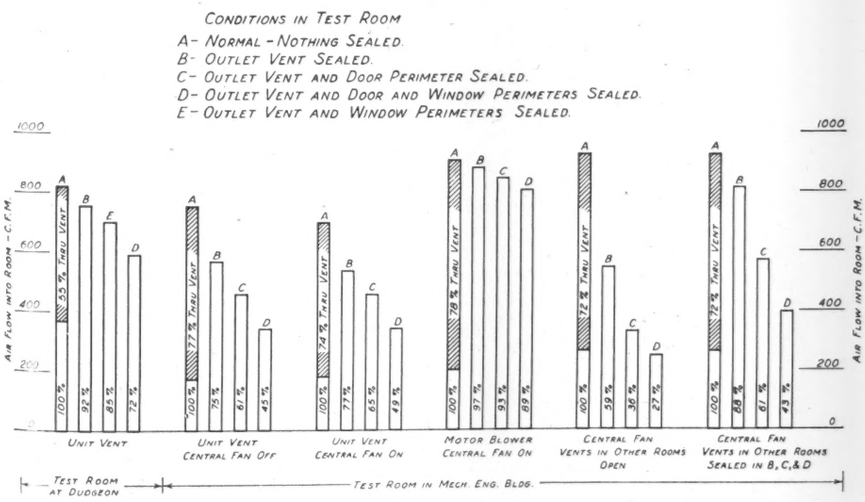

CONDITIONS IN TEST ROOM
A- NORMAL - NOTHING SEALED.
B- OUTLET VENT SEALED.
C- OUTLET VENT AND DOOR PERIMETER SEALED.
D- OUTLET VENT AND DOOR AND WINDOW PERIMETERS SEALED.
E- OUTLET VENT AND WINDOW PERIMETERS SEALED.
AIR FLOW INTO ROOM - C.F.M.
1000
800
600
400
200
0
55 % THRU VENT
100 % 92 % 85 % 72 %
77 % THRU VENT
100 % 75 % 61 % 45 %
74 % THRU VENT
100 % 77 % 65 % 49 %
78 % THRU VENT
100 % 97 % 93 % 89 %
72 % THRU VENT
100 % 59 % 36 % 27 %
72 % THRU VENT
100 % 88 % 61 % 43 %
UNIT VENT
UNIT VENT CENTRAL FAN OFF
UNIT VENT CENTRAL FAN ON
MOTOR BLOWER CENTRAL FAN ON
CENTRAL FAN VENTS IN OTHER ROOMS OPEN
CENTRAL FAN VENTS IN OTHER ROOMS SEALED IN B, C, & D
TEST ROOM AT DUDGEON
TEST ROOM IN MECH. ENG. BLDG.

FIG. 15—SUMMARY OF AIR SUPPLIES UNDER DIFFERENT OUTLET CONDITIONS

obtained in the class room with an eastern exposure with a southeast wind having a velocity of 15 mph and a temperature of 46 F. The check tests were made with a 15 mph west wind and an outside temperature of 50 F. The points determined a curve somewhat lower than shown on Fig. 12 for the central fan with louvers open. The results of the check test with the louvers open made the same day were comparable to the results of the tests with the louvers closed in the other class rooms. The amount passing through the outlet vents seems to be considerably influenced by air infiltration.

Summary of Outlet Vent Study with Mechanical Ventilation

Fig. 15 shows a graphical summary of the influence of sealing various exits on the air supplied by the various systems. The maximum deliveries of the unit ventilators have been used, and air volumes in the unsealed runs chosen for the central fan and motor-blower between 900 cfm and 1000 cfm. The differences in the heights of the normal or *nothing sealed* volumes have no particular significance. The motor-blower and central fan inlet were both capable of supplying considerably more than the quantities shown, but values were chosen close to the range of the unit ventilators tested.

The absence of outlet vents in the Dudgeon School room reduced the capacity only to 92 per cent, whereas this resulted in a reduction to 75 to 77 per cent with the unit ventilator in the Mechanical Engineering Building class room. The central fan, whether it was on or off, exerted little influence on the air delivered by the unit ventilator. This was due to the relatively free discharge of air from the corridor through roof ventilators and the consequent low pressure built up there. Measurements of the pressure drop from the corridor to the outside with an outside temperature of 60 F were 0.009 in. water with the fan off and 0.015 in. water with the fan running.

The greater decrease resulting from the sealing of the outlet vents in the Mechanical Engineering Building than in Dudgeon was due to the newer and better building construction offering more resistance to air passage. The complete sealing of the outlet vent, door and window perimeters at Dudgeon reduced the capacity to 72 per cent and at the Mechanical Engineering Building to 45 to 49 per cent. In this same Mechanical Engineering Building the motor-blower delivered 97 per cent of its unsealed capacity when the outlet vents were sealed and when in addition door and window perimeters were sealed the delivery was still 89 per cent.

With the outlet vents in all other class rooms open as was the case of the previously-mentioned comparison from Fig. 15, the air delivery with the central fan was reduced to 59 per cent upon sealing the outlet vents in the test room. This large reduction is due to the diverting of air to other rooms with the increase of resistance in the test room. With the opening of windows in a particular room, air is diverted from other rooms to the one with these additional outlets. When this is done in many rooms at once, the fan discharge pressure may decrease enough to overload the fan motor.

A truer comparison of a central fan system designed with and without planned outlet vents is given in that

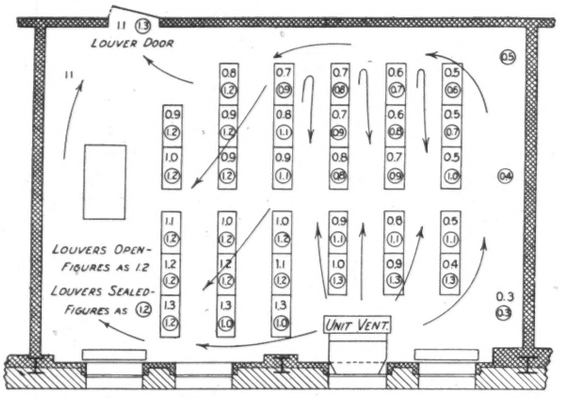


FIG. 16—AIR DISTRIBUTION TESTS WITH UNIT VENTILATOR

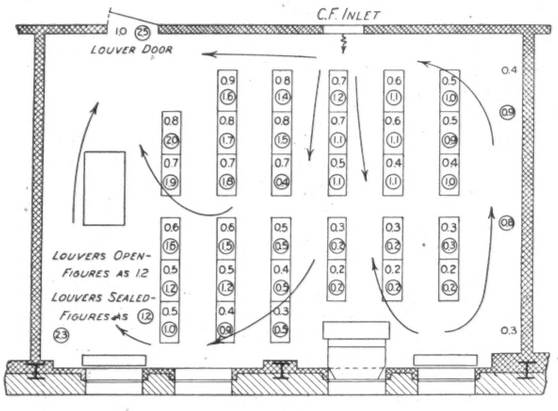


FIG. 17—AIR DISTRIBUTION TESTS WITH CENTRAL FAN

part of Fig. 15 at the extreme right. The correct comparison should be that of the delivery when the outlet vents are open in the test room as well as in all other ventilated rooms with the delivery obtained when the outlet vents in the test room and all other ventilated rooms are sealed. This prevents the diverting of air because the resistance is built up in all rooms at the same time. Under these conditions, sealing of the outlet vents reduced the air delivery to the test room to 88 per cent, and reduced the total delivery to all of the rooms to 86 per cent. This reduction is less than that with the unit ventilators and more than that with the motor-blower. In the further sealing of window and door perimeters, the same procedure should have been carried on in all other class rooms as in the test room. This seemed impractical to do, consequently the 61 per cent for additional sealing of doors and the 43 per cent for the further sealing of windows are a little lower than they would have been with the more exact procedure. With that procedure, it is to be expected that these percentages would be between those for the unit ventilator and the motor-blower tests, whereas with the procedure used, they are very near to the unit ventilator figures.

The slope of the heights of the air volumes resulting from increased sealings of air exits gives an indication of the tightness of a room. The steeper the slope, the tighter the room.

The air passing out of the room through the outlet

vent was 55 per cent of the supply at Dudgeon School. In the various tests at the Mechanical Engineering Building, it varied from 72 to 78 per cent. The larger quantity leaving in the latter building is accounted for in the area of the vent being greater, by its being a door louver opening instead of a vent duct with two 90 deg bends, and the tighter construction of the rooms. As stated in connection with Fig. 14, wind velocity and air infiltration have considerable influence on the amount of air passing through the outlet vents.

Air Distribution in the Class Room

Observations were made on the air distribution with and without planned outlet vents for both the unit ventilator and the central fan. Tests were first made with hydrogen sulphide introduced into the air supply and filter papers dipped in a lead acetate solution placed on 16-in. stands on each of the student tables. The filter paper turned black upon exposure to hydrogen sulphide. The tests were repeated with sulphur bombs which gave off a dense yellow smoke. The smoke method proved to be more satisfactory. The time to reach a given station was longer for the hydrogen sulphide method, and it was difficult to judge the first shade of darkening. Fig. 16 shows the results of the two sulphur bomb tests with the unit ventilator, and Fig. 17 the results for this method with the central fan. The time in minutes to reach each station 4 ft above the floor for the sealed louver runs are shown circled and are shown plain for the unsealed louver runs.

The unit ventilator volumes were 700 cfm in the unsealed and 540 cfm in the sealed run. The corresponding deliveries for the central fan runs were 1080 cfm and 500 cfm.

The distribution seemed very good in every test. There seemed to be no shorting to the outlet vent when unsealed and no undue sluggishness in reaching all locations when the outlet vents were sealed. In every case most of the air travelled across the top of the room from the inlet before dropping to the 4-ft level. The shortest time recorded to a table station with the outlet vent open and with the unit ventilator supply was 0.4 min, and the longest was 1.3 min. With the outlet vent sealed the shortest time was 0.6 min, and the longest 1.3 min. With the central fan system and the outlet vent open, these times were 0.2 min and 0.9 min. With this system and the outlet vent sealed, these times were 0.2 min and 2.0 min. The time to reach the louver opening in the door in the unsealed tests was practically one minute in each case. In the sealed test with the unit ventilator, the time to reach the door was 1.3 min, and was 2.5 min with the central fan system. The ½-in. crack under the door offered the definite exit for air at this location when the louver opening was sealed. It is considered that the distribution was very good in each case and that no evidence is shown in Figs. 16 and 17 for or against planned outlet vents.

Open Window Ventilation

Observations have been taken over a period of several years on open window ventilation in two typical class rooms in the main Engineering Building. Two rooms on the second floor were selected, one with a north exposure and one across the corridor with a south exposure. The building is a 4-story structure with a full ground floor so that the rooms selected were considered to be near mid-height and the stack effect as low as it would be at any place in the building.

Two series of observations have been made, one with the outlet vents closed and the other with the outlet vents open. The building is ventilated by a central fan system. The inlet grilles to the two rooms were kept sealed at all times. The south room was 20 by 30 ft in size and had five windows approximately 3 ft 8 in.

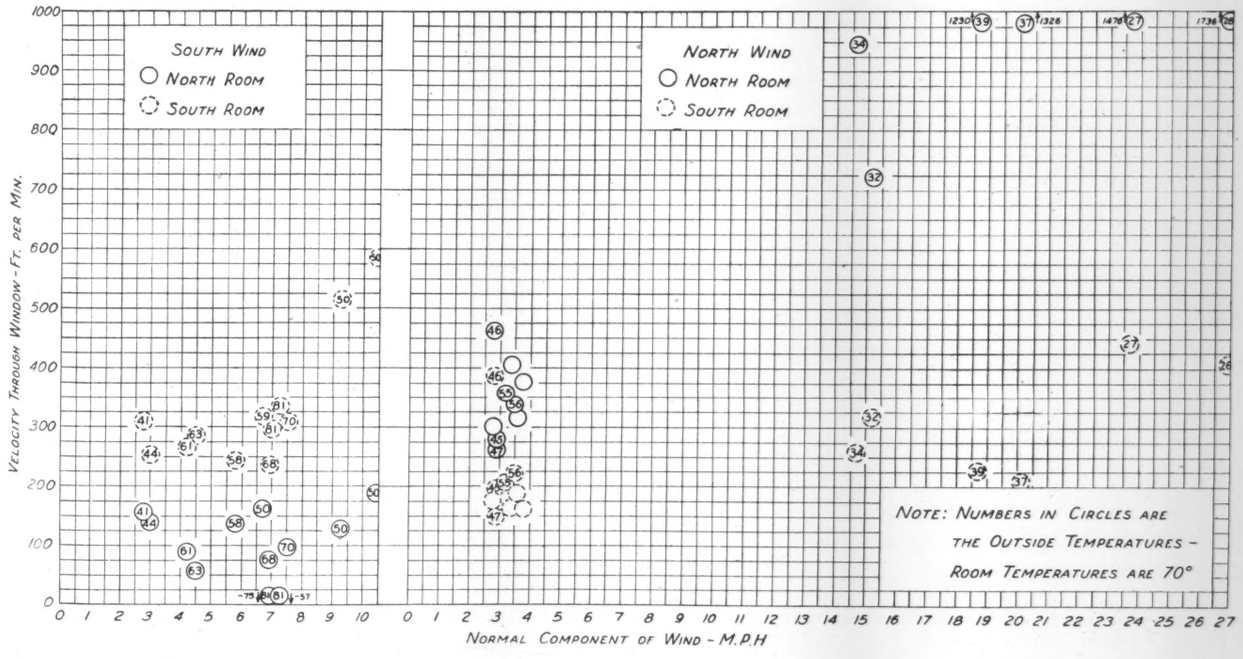

South Wind
North Room
South Room
North Wind
North Room
South Room
Note: Numbers in Circles are the Outside Temperatures - Room Temperatures are 70°
Velocity Through Window - Ft. per Min.
Normal Component of Wind - M.P.H.

Fig. 18—Open Window Ventilation With Outlet Vents Open

by 7 ft 0 in. The outlet grille was 27 in. by 29 in. The north room dimensions were 24 by 26 ft and there were three windows of the same size as in the south room. The outlet grille was also 27 in. by 29 in. Fig. 3 shows a view in the south room, showing an open window and the outlet vent.

The procedure in testing was to open one window in each room a definite height as would be used for open window ventilation. This height was established at 4½ in. An anemometer was set in the center of each window opening and readings observed at 5-min intervals. At the end of one-half hour, anemometers were exchanged between the two rooms and readings continued another one-half hour. This was considered advisable to eliminate any difference in anemometers under variable flow conditions, especially at low velocities. Corrections were applied to the readings, but at very low velocities the readings would not be accurate.

The wind velocity and direction and the outside temperature during the hour of observations were determined from the charts of the local weather station located on the roof of the next building. The charts give a continuous record of the wind velocity and the direction for each minute. Observations were made for winds with a general direction of south and north. In using the results, the one-half hour averages of the velocities through the open window were plotted against the component of the wind that was normal to the building wall. This was obtained by correcting the individual one minute readings of velocity so as to get the component perpendicular to the wall. These values were then averaged for the one-half hour periods.

Fig. 18 shows the results of the tests with the outlet vents open and Fig. 19 shows the results of the tests with the outlet vents sealed. The results for the south winds are shown on the left and for the north winds on the right of these figures. Although the mid-height of the building was chosen to lessen the effect of stack action from temperature differences, temperatures affected the results considerably. The outside temperatures are consequently shown within the circles representing the data points. The north room results are shown as solid circles and the south room results as dotted circles in each case.

The results show that under almost every observed condition there was a considerable inflow of air in both the windward and leeward rooms. Only in one case was an outflow found in even the leeward room and that was with an outside temperature of 81 F and a wind velocity of about 7 mph. The air came into the room through the open window with a velocity ranging usually from 200 to 500 fpm. The area of the window opening was 1.37 sq ft. In calibrating an anemometer in an opening made to simulate an open window, the coefficient of several runs at different rates of flow was found to be within one per cent of 0.74. The air flow in cfm through the opening used would then be about the same as the velocity in feet per minute.

It is difficult to compare the effect of outlet vents by studying Figs. 18 and 19 because of the effect of outside temperatures. For instance in Fig. 18 for a south wind with outlet vents the points lie lower than in Fig. 19 for the same conditions except without outlet vents. This is due to the higher outside temperatures during the test with outlet vents. At high wind velocities the influence of the outlet vents is noticeable. For instance, with a north wind having a velocity of 21 mph, the velocity of inflow in the windward room according to Fig. 19 would be 975 fpm without outlet vents at an outside temperature of 27 F. Fig. 18 shows that with outlet vents, the inflow was at the rate of 1476 cfm at the same temperature, but a wind velocity of 23.6 mph.

In order to trace the effect of outlet vents on the inflow with open window ventilation an attempt was made

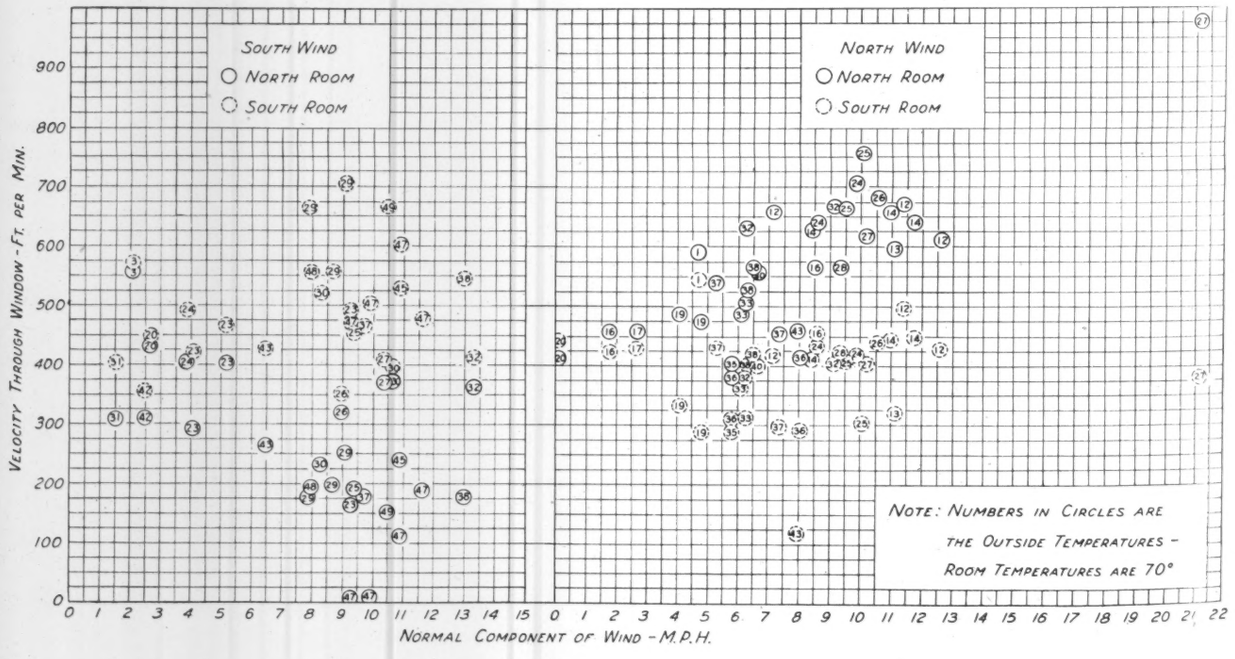

South Wind
North Room
South Room
North Wind
North Room
South Room
Note: Numbers in Circles are the Outside Temperatures - Room Temperatures are 70°
Velocity Through Window - Ft. per Min.
Normal Component of Wind - M.P.H.

Fig. 19—Open Window Ventilation With Outlet Vents Closed

to eliminate the temperature effect. The assumptions were made that the wind has no effect on the inflow on the leeward side and that the difference in inflow on the windward and leeward rooms was due to the normal component of the wind velocity. The differences in inflow were found for both north and south winds with and without outlet vents. These differences were then divided by the wind velocity so that a figure was obtained that represented the inflow velocity due to wind effect in feet per minute per mile of wind velocity.

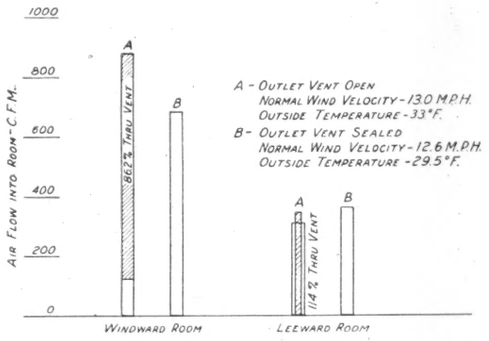


FIG. 20—OUTLET VENT INFLUENCE ON OPEN WINDOW VENTILATION

The average found for the windward room with a south wind and with the outlet vents open was 38.8 fpm per mile of wind velocity. The corresponding average for a north wind, the north room now being the windward room, was 40.1 fpm per mile of wind velocity. With the outlet vents sealed the average found with a south wind was 29.8 fpm per mile of wind velocity and with a north wind the average was 24.3 fpm per mile of wind velocity. Most of the tests with the outlet vents sealed were made after a small structure was erected on the north side of the building that sheltered to some extent the test window from northwest winds. In obtaining the average of 24.3 fpm the points with a northwest wind were eliminated and later tests were made only with north and northeast winds. The average with the faulty points included was 21.2 fpm per mile of wind velocity.

The reduction in per cent for the sealing of the outlet vents was 26 for the north wind and 37 for the south wind. This is an average for both of a reduction to 69 per cent for the wind effect on open window ventilation with the elimination of outlet vents. The corresponding figure for the unit ventilator installation in the Mechanical Engineering Building was 75 to 77 per cent.

Fig. 20 shows another study of the effect of outlet vents with open window ventilation in the same two class rooms in the main Engineering Building. Tests were made with the outlet vents open and directly following with the outlet vents closed. In addition to the measuring of the air entering the rooms through the open windows, the air leaving through the outlet vents was measured when they were not sealed.

With a normal wind velocity of 13 mph, 880 cfm entered the windward room and 320 cfm entered the leeward room. In the windward room, 86 per cent as much air left by the outlet vent as entered the open window. In the leeward room, 14 per cent more air left by the outlet vent than entered by the open window. This shows the drift of air from the windward side to the outlet vents of the leeward side. With the sealing of the outlet vents the inflow on the windward side was reduced to 690 cfm. On the leeward side the air flow increased to 360 cfm. The reduction on the windward side amounted to 22 per cent. The only accounting for the increase in inflow on the leeward side of 50 cfm was in the slight reduction of 0.4 mph in wind velocity and the lowering of outside temperature of 3.5 deg.

General Summary

The amount of air passing out through the planned outlet vents of a class room in per cent of the air supply was found to be 55 for a building of comparatively poor construction in so far as tightness to air leakage is concerned. In a new building of very good construction, this per cent was found to be 75. This is not a fixed percentage of the planned air supply but rather is dependent considerably on infiltration. In one series of tests, the air leaving by the planned outlet vent was over 100 per cent of the planned air supply.

The pressures built up were not sufficient to overcome infiltration at ordinary wind velocities. With the planned outlet vent sealed, more than 600 cfm could be supplied to the room by the fan at a pressure equal to that of a 15 mph wind.

Sealing of the planned outlet vents resulted in a reduction in air delivery to the room of from 3 to 25 per cent of the air delivery with the outlet vents open. This percentage depends on the fan characteristics as to the ability to maintain volume of air delivery as the static pressure builds up.

The study of open window ventilation shows an inflow on both the windward and leeward sides of some 200 to 500 fpm under most wind velocities and outside temperatures observed. The inflow was almost 1500 fpm in one case of high wind velocity.

Although rooms were selected near the mid-height of the building, the temperature effect had an influence on the inflow of air through the open windows. In order to obtain a figure for wind effect only, the assumption was made that the difference in inflow in the windward and leeward rooms was due to the normal component of the wind velocity. On this basis, an average of slightly less than 40 fpm resulted from each mile of wind velocity when the outlet vents were open and 27 fpm resulted when the outlet vents were sealed.

Greenhouse Heating

In greenhouse heating, it is important that the heat be diffused evenly, that there be no high-velocity air currents, especially for flowers, and that no rapid temperature fluctuation be permitted.

Steam and hot water systems both are used in greenhouse heating. Electrically-propelled unit heaters have been used with good results.

The heat losses from greenhouses are computed the same as are the heat losses from other types of buildings.

Automatic Gas Burners

By C. George Segeler[1], *(NON-MEMBER)*
New York, N. Y.

THE application of gas can be made through two general methods, namely (1) by the use of equipment specially designed for gas burning, and (2) by installing gas burners in existing equipment intended for some other fuel. Considerable information has been published on the use of gas with specially-designed equipment, but there has been comparatively little on the automatic gas burner used for converting existing equipment to gas operation.

The units for accomplishing this purpose are termed *conversion burners*. Since these burners are really gas burners fitted with the same equipment and controls used with boilers or furnaces designed for gas operation, it is to be regretted that the term *conversion burner* has found acceptance.

The published papers on automatic gas burners have generally been written from the standpoint of the operating cost. Although it is true that greater care is necessary in the selection of conversion burners where the gas cost is high, it is also true that automatic gas burners can and are being successfully used in territories in which gas-designed equipment is used. In the past, statements to the effect that automatic gas burners could be made as efficient as gas-designed equipment have often been questioned, but as the author will endeavor to show later, this is sometimes the case if sufficient care is exercised in selecting the jobs converted and proper installation and adjustment of the burner is made afterwards.

Where automatic gas burners are to be used in existing buildings, the installed equipment must be adequate, it must not be too old, and it must be of a type for which a fairly efficient conversion can be made.

Although there are exceptions, small boilers usually are less satisfactory than large boilers for conversion to gas operation. This is particularly true of small round boilers without intermediate sections. Such a statement in no way reflects upon any manufacturer of coal-fired boilers. It does not indicate that such a boiler is not efficient when used with coal, but merely that the construction is such that it is not well adapted for gas.

Coal-fired, warm-air furnaces can be divided into three classes, namely (1) self-cleaning dome type, (2) radiator dome type and (3) downdraft type. The latter two types can be changed to automatic gas operation satisfactorily, but considerable care must be exercised in converting the first type.

Lest any misunderstanding arise from the foregoing reference to the difficulties attendant upon adapting the automatic gas burner to the small coal boiler, it should be noted that there are also other factors than size which may limit the use of gas burners. Space will not permit enumerating all of them, but the principal points are as follows:

1. There is a difference in boiler construction between units intended for anthracite and bituminous coal, principally based on draft consideration.
2. Age of the boiler, especially when hard water is used.
3. Capacity of the boiler compared to the heating requirements of the house.
4. The chimney may be a factor, since flues too small for good coal operation may often be entirely satisfactory for oil or gas burning. Outside exposed chimneys with unlined flues are liable to be unsatisfactory.

Having more or less defined the field of usefulness of the automatic gas burner, the differences between the individual burners themselves can be considered. Automatic gas burners may be simply classified into the following three groups: (1) elementary atmospheric burners, (2) burners with secondary air control, and (3) burners with air blowers.

Elementary Atmospheric Burners

The simplest burners consist of plain drilled pipe set into the firebox of the coal boiler. No attempt is made to control the amount of excess air which enters the furnace and no refractories are used. Burners of this class are made by many local concerns throughout the natural gas territory. They are obviously inefficient and undesirable from a service angle.

Special shapes of atmospheric type burners are also used, such as the single port burners which are set into the fire door and play a large volume of flame from a single port. The nozzle-mixing multi-port burners, also of this class, are widely used not only for domestic installations but for the conversion of the larger size boilers. For the latter, this class has certain specific advantages in simplicity and long life, the details of which need not be considered in this paper.

Burners with Secondary Air Control

Because many of the installations made with elementary atmospheric burners yielded rather low efficiencies, secondary air-controlled types of automatic gas burners were developed. These are arranged to admit all air for combustion through a sheet metal duct surrounding the burner. An air control damper opens and closes with the position of the gas valve, thereby controlling the secondary air during the *on* period, and shutting off the air flow during the *off* periods.

[1] Industrial Engineer, *American Gas Association.*

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Milwaukee, Wis., June 1932.

Burners with Air Blowers

The third broad class of burners are those using an air blower for the purpose of securing improved combustion and better control. These units do not operate with 100 per cent primary air mixed with the gas by the blower. This may be surprising to many, since it is frequently assumed that the blower system would inject all of the required air with the gas. This is not usually done because of the nature of gas combustion. In order to obtain silent combustion and freedom from flashback, it is desirable in house heating equipment to mix about 60 per cent of the air required as primary air.

Practically all automatic burners are designed to locate the flames in close proximity to the heating surface. This has been found to give a relatively high efficiency, and makes the lighting of the burners very positive. However, there are certain exceptions to this construction, notably in the elementary atmospheric types of burners and in the burners designed to operate with special refractory shapes in the center of the firebox.

There is another type of burner which does not fall exactly into any of the classifications mentioned. In this class there is no air injection external to the boiler. The burners are designed from small bunsen-type tubes, the air and gas mixing taking place within the combustion space itself.

With so many different types and styles available, the question arises as to which should be selected. There seem to be four principal factors involved. *First,* that type of burner should be selected which would require the smallest investment consistent with safety, efficiency and satisfactory service. *Second,* the burner selected should cause the least possible change in existing equipment, so that the owner would not feel that he was permanently bound to gas fuel and so that the working principle of the unit would be most readily understood. *Third,* the unit should possess a neat appearance when installed and should have a low cost for installation and above all, a low cost for servicing. *Fourth,* the unit should be such that the owner receives an efficient and safe device.

Installation of Automatic Gas Burners

Considerable experience is required to satisfactorily install an automatic gas burner in existing equipment. With gas-designed equipment, it is merely necessary to follow instructions for assembly to be sure of good operation. This is well within the ability of any capable mechanic. For the installation of automatic burners, on the other hand, it is not only a question of following certain instruction rules, but it is also necessary to have a mechanic with sufficient knowledge of gas combustion and heating methods to be able to make the necessary changes in the original equipment so that with the burner it will make an efficient and safe installation.

The *American Gas Association* Testing Laboratory Seal of Approval which is given on appliances that meet the specifications of the *A. G. A.* General Requirements Committee[2] cannot be extended to cover automatic gas burners because, although the gas-designed unit can be accurately assembled by any competent mechanic so that it will comply with the requirements under which it was tested for approval, the automatic gas burner depends upon the skill of the installer. Consequently, it would be impossible to issue a Seal of Approval, which bears the label, *Complies with basic national standards for safety.*

In order that the automatic gas burner might be given consideration, the *A. G. A.* General Requirements Committee has approved the issuance of the Installation Requirements for Conversion Burners. These are requirements which, when followed, will make the finished installation comparable with gas-designed equipment as far as it is possible to do so with this equipment. These requirements are divided into several parts, covering the inspection and preparation of the existing heating plant, with particular reference to chimney flues and proper size of flue pipe. They provide that no baffles shall be installed in existing equipment which will interfere with proper combustion of gas, and recommend that the section of flue pipe between the outlet of the appliance and the draft hood shall be sized on the basis of 1 sq in. of area for each 7500 Btu per hour input into the unit. The second part deals with the actual installation of burners and controls. In general, the same type and number of controls are needed for automatic gas burners as for gas-designed equipment. This includes pilots, main shut-off valves, gas pressure regulator, necessary limiting devices, etc. The third part deals with the gas piping and meter arrangement, and a final section is devoted to inspections and test. This material is available on application from the *American Gas Association.*

The Requirements represent minimum standards for the installation of automatic gas burners. They have been prepared in the interest of the public, to ensure safe and economical utilization of gas. Naturally, there are companies whose practices will go beyond these minimum standards and will thus be able to reach the highest levels of efficiency and good service. Most of the recommendations in the Installation Requirements are sufficiently definite that they require practically no further interpretation. There are some which have been deliberately left general, because of variations in local practice.

Most important of these is the method for sizing automatic gas burners to fit a given installation. The *A. G. A.* practice is based on first estimating the hourly heat loss of the house, and then providing a sufficiently high heat input per hour to the unit to take care of the following three points:

1. Efficiency of the unit.
2. Heat lost from risers, piping and leaders and thus not delivered to the rooms.
3. Extra capacity for quick starting under thermostatic control.

For warm air furnaces, it is assumed that 90 per cent of the heat delivered to the heat distributing pipes is finally brought into the rooms of the house; provided that where leaders pass through unexcavated spaces, they are insulated and that the area of the air passages through the furnace is at least equal to the combined area of the take-off pipes.

[2] This committee consists of representatives of U. S. Bureau of Standards, U. S. Bureau of Mines, U. S. Public Health Service, U. S. Bureau of Home Economics, *American Home Economics Association,* Underwriters Laboratories, *Heating and Piping Contractors National Association, Canadian Gas Association* and gas company and manufacturers' members of the *A. G. A.*

Combining the probable efficiency of the automatic gas burner with these requirements, the input for warm air furnace conversions can be based on the following rule: *Multiply the hourly Btu loss of the house by* 1.56. *The answer will be the required hourly input for automatic burners for warm-air furnaces.*

The *A. G. A.* Requirements for gas-designed, warm-air furnaces call for gas-tightness between the heating section and the air passages. A similar criterion should be applied to existing warm air furnaces before they are accepted for conversion to gas firing. If any doubt exists about the tightness of the joints in the heating surfaces, it should not be difficult to make a wintergreen or smudge test to make certain that the furnace joints are actually tight.

Boilers are sized in a similar manner. The *A. G. A.* long ago adopted a standard practice of sizing boilers up to 500 sq ft of installed equivalent heating surface (radiation) in the house on a basis of 56 per cent more boiler capacity than standing heating surface. The excess capacity decreases up to 4,000 sq ft of installed heating surface for which the figure is 40 per cent. In dealing with automatic gas burners, it has been suggested that a slightly smaller excess capacity can be satisfactorily used. The sizing can be done by multiplying the hourly heat loss of the house by the factors given in Table 1.

In converting large boilers it is advisable to work out the details of the installation rather than to depend on tabular factors for estimating the gas required. Continuously-operating boilers, boilers for factories, clubs, etc., may have special requirements for quick pick-up, or the piping arrangements may be such that empirical factors are not as satisfactory. However, it is not possible to cover the details of such situations in the short space of this paper. The general rule can be applied that the excess capacity may be reduced as the boiler size increases.

It has been suggested that the proper operating efficiency for automatic gas burners be obtained by depending upon a flue collar or properly-sized flue outlet to provide the necessary baffling effect rather than using baffles or restrictions in the flue passages of the equipment. The safe basis for operation is 7500 Btu input per hour per square inch of flue opening, and an oxygen content in the flue gases of not less than 4 per cent nor more than 10 per cent. Such a requirement does not necessarily provide the lowest obtainable flue temperatures. It should be readily possible, however, to bring these down to between 250 F to 350 F for steam boilers.

TABLE 1—FACTORS FOR SIZING BOILERS

INSTALLED HEATING SURFACE (SQUARE FEET)		FACTOR	FOR CALCULATING REQUIRED GAS INPUT, MULTIPLY HOURLY BTU LOSS BY (75% EFFICIENCY ASSUMED)
STEAM	HOT WATER		
200	350	1.58	2.11
250	400	1.57	2.09
300	470	1.53	2.04
400	640	1.50	2.00
500	800	1.46	1.95
600	1000	1.42	1.90
700	1100	1.38	1.83

These are relatively broad requirements and some, interested in securing higher thermal efficiencies, would want to impose further restrictions. For example, by using a smaller flue opening and reducing the oxygen content of the flue gases to the minimum figure, the thermal efficiency can be increased slightly. According to the principles of operating gas appliances, there must be no carbon monoxide in the flue gases at any time. An ordinary flue gas analysis by the Orsat method will not be sufficiently accurate to use as a test of compliance with the requirements for absence of carbon monoxide. In coal or oil-burning appliances, where unbroken flue connections are used, it would not be practical, and probably would be unnecessary, to attempt to restrict the carbon monoxide content of the flue gases to the same degree required for gas-burning appliances. The limit of carbon monoxide permissible for gas-designed central heating appliances is not more than 0.04 per cent in an air-free sample of the flue products. There are portable types of carbon monoxide indicators and detectors which give very quick determinations of carbon monoxide without going through the complications of an Orsat analysis. By such methods, efficiencies can be brought to high levels. In fact, under the conditions of a series of tests run by a well-known university laboratory, the efficiency of converted systems was found to be on a par with that of gas-designed equipment.

On the other hand, there are cases, especially in territories where natural gas is available at a low rate, in which it is not necessary to make an accurate heat loss estimate of the house to check the installed heating surface to test converted systems, simply because the savings in fuel resulting do not warrant it.

In order to obtain additional data along the lines developed by the university laboratory already referred to, an investigation was started among the larger gas companies, obtaining from each a list of typical steam and hot water installations. This list, although a random selection, was picked to have approximately the same number and size installations in the following four classes:

1. Gas-designed hot water boilers.
2. Converted hot water boilers.
3. Gas-designed steam boilers.
4. Converted steam boilers.

Thirty-two gas companies participated in sending in data, and more than 700 jobs were listed. A very excellent coverage of the field was thus obtained.

The data have been reported on the basis of Btu of gas required per degree day per 1000 Btu hourly heat loss from the building. Theoretically, a unit figure can readily be calculated corresponding to 100 per cent thermal efficiency as follows:

> Assume that a house having an hourly heat loss of 100,000 Btu, designed for an outside temperature of 0 deg F and an inside temperature of 70 F, were operated continuously through a period of zero days. That is to say, the house would lose
>
> $$100{,}000 \times 24 = 2{,}400{,}000 \text{ Btu}$$
>
> each day. On a zero day there are 65 degree days; consequently, the amount of fuel required to heat this house on a 100 per cent efficiency basis would be 370 Btu per degree day per 1000 Btu hourly loss from the house.

Fig. 1 illustrates the ratio of output to input of houses according to the methods used in this paper. The

typical example tabulated at the left of the illustration assumes a house having a total heat loss of 100,000 Btu per hour, and divides this into losses from infiltration through glass and doors, walls, roof and floor. On the right of the illustration are tabulated the results of a survey of gas-fired houses made in 1930 by the author in an attempt to ascertain the percentage of heat lost through the various parts of the house. The houses were measured by the methods given in the A. S. H. V. E. GUIDE, and the percentages tabulated and averaged. The input in Btu delivered by the gas is given for various seasonal efficiencies on the basis of units of 1000 Btu loss per hour from the building per degree day.

The actual results obtained from the present survey of 700 private residences averaged as follows:

Type of Installation	*Btu per degree day per 1000 Btu hourly heat loss from the house*
Gas-designed hot water boilers	404
Gas-designed steam boilers	445
Converted hot water boilers	452
Converted steam boilers	503

These figures show overall seasonal heating efficiencies from a maximum of 91.6 per cent to a minimum of 73.5 per cent. They also show that the overall results of hot water equipment are 10 per cent better than steam equipment and that the difference between the average efficiency of all gas-designed versus converted equipment was about 12 per cent in favor of the gas-designed equipment.

Fig. 2 shows the relationship between the size of the installation and the average gas consumption, expressed in Btu of gas required for each 1,000 Btu hourly heat loss from the building, figured in accordance with the methods given in the A. S. H. V. E. GUIDE, per degree day. This figure refers to the results obtained for gas-designed hot water boilers. These data are all for private residences. As the size of the job increases the unit gas requirement seems to decrease.

Fig. 3 is a similar plot based on gas-designed steam boilers. Fig. 4 is another similar plot based on converted hot water boilers.

In all of these three figures, it appears that larger installations were more efficient than smaller ones. Perhaps there is some justification for this interpretation, but there are some other points to consider. For the moment, it may be assumed that these charts represent the actual averages of a number of installations in the various size groups plotted.

When the lists of jobs were studied in detail in any one size class, it was found that individual installations showed extremely wide differences in efficiency. It showed that there was a small group of gas-designed jobs that were better than the best of the conversions and it also showed that a small group of converted installations was less efficient than the least efficient of the gas-designed equipment. The majority of the jobs was on a par as far as efficiency is concerned; the 12 per cent difference in average efficiency was largely caused by the installations which fell into the highest and lowest groups.

Fig. 5 illustrates the range of values obtained in this study. It can readily be seen how the averages worked out favorably to the gas-designed equipment even though there was a large number of installations in each class which yielded overlapping unit figures. Eliminating freaks, the worst of the converted installations averaged about 558 Btu per degree day per 1000 Btu hourly loss

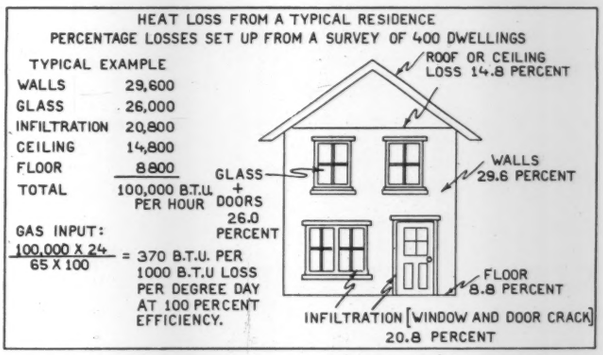

HEAT LOSS FROM A TYPICAL RESIDENCE
PERCENTAGE LOSSES SET UP FROM A SURVEY OF 400 DWELLINGS
TYPICAL EXAMPLE
WALLS 29,600
GLASS 26,000
INFILTRATION 20,800
CEILING 14,800
FLOOR 8800
TOTAL 100,000 B.T.U. PER HOUR
GAS INPUT:
100,000 X 24 / 65 X 100 = 370 B.T.U. PER 1000 B.T.U LOSS PER DEGREE DAY AT 100 PERCENT EFFICIENCY.
ROOF OR CEILING LOSS 14.8 PERCENT
WALLS 29.6 PERCENT
GLASS + DOORS 26.0 PERCENT
FLOOR 8.8 PERCENT
INFILTRATION [WINDOW AND DOOR CRACK] 20.8 PERCENT

FIG. 1—RATIO OF OUTPUT TO INPUT

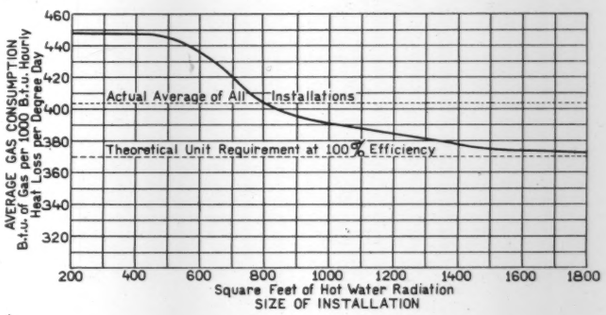


FIG. 2—UNIT GAS REQUIREMENTS FOR HEATING RESIDENCES WITH GAS-DESIGNED HOT WATER BOILERS

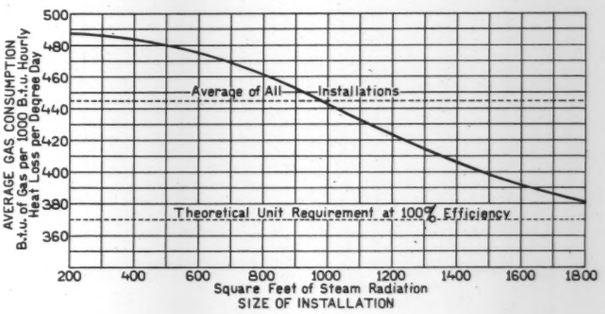


FIG. 3—UNIT GAS REQUIREMENTS FOR HEATING RESIDENCES WITH GAS-DESIGNED STEAM BOILERS

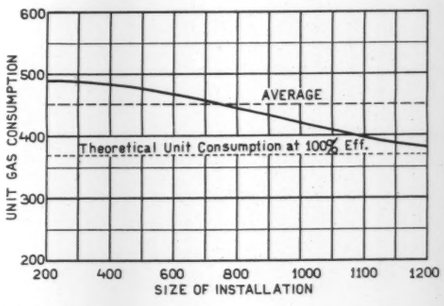


FIG. 4—UNIT GAS REQUIREMENTS FOR HEATING RESIDENCES WITH AUTOMATIC GAS BURNERS IN WATER BOILERS

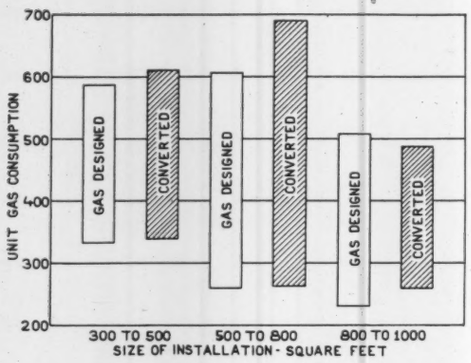


FIG. 5—RANGE OF UNIT GAS CONSUMPTIONS FOR GAS-DESIGNED AND CONVERTED HOT WATER BOILERS

from the building, as contrasted with 525 for the worst of the gas-designed.

The extraordinary part of these data was the unit figures obtained from the best of the installations. The best of the gas-designed jobs show about 250 Btu per degree day per 1000 Btu hourly heat loss from the building, and the best of the converted installations about 274 Btu. Taking the two unit figures just mentioned as an example, they indicate that the overall seasonal thermal efficiencies were 140 per cent and 150 per cent respectively. In other words, the theoretical heat losses, calculated according to recognized principles, and making due allowance for every factor which was known to be involved, were found to be one and one-half times as great as the heat input furnished by the gas.

The practice followed by the engineers of the gas companies who contributed the data used in this paper is to draw up complete data sheets showing the hourly heat loss of each house. These calculations are made strictly in accordance with the methods developed by the Society and published in THE GUIDE. The fuel consumption is metered separately from the other uses for gas in these houses, and it is reasonable to assume that the meter accuracy is better than ± 2 per cent. The Btu value of gas, although it fluctuates somewhat, is averaged by the gas companies and it was these monthly average figures of Btu input which were correlated with the heat loss of the house. It was expected that a reasonably close check could be obtained between the input in gas Btu and the seasonal heat loss of the house.

So much was expected, but actually the results turned out as has been indicated, namely, that there was no very close check between the input and the output of houses, and that it was unwise to attempt to employ heat loss calculations on buildings as a basis for estimating the seasonal efficiency of various heating systems.

If a close check had been obtained between the maximum and minimum unit figures, or between the input and output estimates of houses, it would point out the value and desirability of refining still further the excellent methods for calculating and estimating heat losses and heat requirements as outlined in THE GUIDE and in the many valuable research papers published by the Society based on results obtained at the A. S. H. V. E. Research Laboratory and at the cooperating institutions. The results obtained did not show a close check, but on the contrary a wide divergence; high and low values were as much as 2 to 1 in relation to one another. Seasonal efficiencies figured out to theoretical results of better than 100 per cent. These factors have raised some question in the minds of the gas engineers on the advisability of carrying out too great a degree of refinement in the calculation of building heat losses and the use of this information.

Referring to the description of Fig. 3 in this paper, the significance of the seeming increase in efficiency of the larger installations may be noted. It is quite possible that this seeming increase in the case of larger houses could be traced to increasing errors which may affect the accuracy of heat loss estimates of the smaller houses.

The total of 700 installations seems sufficient to warrant placing considerable reliance on the general average data which have been deduced from them. Nevertheless this does not imply that such figures possess absolute accuracy, especially since it is well known to the gas industry that the results of houses in southern territories show lower seasonal efficiencies than those in the north.

Society Adopts Four Standard Codes

IN ACCORDANCE with the resolutions adopted by the Society four Codes were submitted to the members for vote by letter ballot as A. S. H. V. E. Standards and on June 15, the Board of Tellers appointed by Pres. F. B. Rowley reported as follows:

Report of Tellers

June 15, 1932

The undersigned tellers report that they have counted and checked all ballots cast and find the following to be a true count:

1. A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand Fired Boilers	FOR	315	
	AGAINST	15	
	NO VOTE	5	
			335
2. A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel	FOR	317	
	AGAINST	11	
	NO VOTE	7	
			335
3. A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)	FOR	316	
	AGAINST	13	
	NO VOTE	6	
			335
4. A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators	FOR	315	
	AGAINST	8	
	NO VOTE	12	
			335

BOARD OF TELLERS,

A. J. Offner, *Chairman*,

M. W. Ehrlich,

C. H. Flink.

Local Chapter Reports

Cleveland

May 4, 1932. At this meeting held in the rooms of the *Cleveland Engineering Society,* Mr. McCutcheon of the Reliance Electric Company outlined more fully the plan sponsored by the *Cleveland Engineering Society* and affiliated organizations to put 1,200 unemployed technical men and engineers at work for the cultivation of approximately 2,000 acres of land, the use of which has been donated by Thomas White. The method of paying the men thus employed, and the manner of reimbursing the friends of the organizations who were willing to advance enough money to get this project started, were described fully by Mr. McCutcheon.

W. C. Kammerer read a report from the Consulting Engineers Division of the Cleveland Engineering Society regarding so-called *bootleg* engineers, that is, salaried employes with other means of support who are doing engineering, usually at cut rate prices, the sum total of which cuts in greatly on the amount of work now available for the legitimate engineer. Pres. W. E. Stark appointed Messrs. Kammerer, Nobis, Morris and Wright to investigate and to report at the next Chapter meeting concerning action to be taken by the Cleveland Chapter beyond the motion made and carried at the February 1932 meeting which discouraged this practice.

Otto Fox and Philip Cohen were unanimously elected members of the Cleveland Chapter. C. F. Eveleth was elected chairman of the Nominating Committee consisting of Mr. Eveleth, W. E. Kammerer and T. A. Weager. This committee was instructed by President Stark to select candidates for the various offices that are to be filled at the June meeting.

The speaker of the evening, Fred C. Belser, Manager of the Central Division of The Ric-Wil Co., gave an illustrated talk on Underground Conduit Construction. He told about the history of The Ric-Wil Co. during its 22 years of experience in Barberton and Cleveland, and showed the changes and progress made in underground conduit construction during that time. The slides showing actual installations emphasized the ease and permanency of the installation of factory-made equipment and indicated the method of welding employed in pipe lines, method of draining the conduits, construction of manholes, and the application of the insulation. Mr. Belser described a test made when required on Government work, where an efficiency of 90 per cent is usually obtained.

In the discussion following the talk, experiences with underground conduit construction were related, some of which, particularly one installation described by Mr. Eveleth, dated back to a time when materials were used that later became patented and standardized.

A vote of thanks was given Mr. Belser and to F. H. Morris as chairman of the Program Committee.

Kansas City

May 16, 1932. An unusual number of members and guests were present at the May meeting of the Kansas City Chapter held on the Roof Garden of the Ambassador Hotel.

The meeting was called to order by Pres. J. M. Arthur, and the routine business included the Treasurer's report, various committee reports, and a discussion of a proposed resolution. A resolution committee was appointed, which returned before the meeting was over with a satisfactory resolution which was voted on and unanimously passed. This resolution has been sent to the Council of the Society with copies to the local Chapters.

Col. Fred Whitten presented, as the civic feature of the meeting, a most forceful, interesting, and inspiring talk on the factors which go to make up the character and integrity of our nation, bringing the subject down finally and forcibly to the individual citizen, as a focal point for constructive action.

Following this most interesting feature, the film, Empires of Steel, was shown depicting step by step from the smelter to the completed structure the fabrication of the Empire State Building, New York City, the world's greatest skyscraper.

This was the last regular business meeting of the year, and was very successful.

Illinois

May 9, 1932. This meeting was held in the Crystal Room of the Hotel Sherman, with 44 present at the dinner and 57 at the meeting. The minutes of the previous meeting were waived by motion.

Eugene Mathis presented the following applications: C. W. Adams, W. A. Russel & Co., A. H. Zimmerman, Chicago Board of Health, W. H. Wilson, The Pullman Co. A cordial welcome was extended to these three new members.

H. G. Thomas, chairman of the Golf Committee, announced that the May golf tournament would be held on May 26 at Wilmette in conjunction with the *Western Trade Golf Association* tournament.

Homer Linn read a letter from G. W. Hubbard, thanking the Chapter for the flowers and expression of friendship during his illness.

Mr. Thomas then read a report from the Committee to Investigate the feasibility of a Better Relations Commission for the heating industry. He announced that the Board of Governors were in favor of the Chapter approving the constitution and by-laws of this new Commission, and wished to place it before the meeting for adoption. He moved that the constitution be adopted as read, but J. F. Hale called attention to the possibility that it might be necessary for the Council of the Society to give its approval before the Chapter could approve. As a result, the motion was amended as follows: It was moved by Mr. Thomas and seconded by Mr. Linn that the Illinois Chapter ratify the proposed constitution and by-laws of the Better Relations Commission of Chicago, and accept membership in the Commission, subject to the approval of the Council of the Society. The motion was unanimously passed. Pres. J. J. Aeberly then appointed a committee consisting of John Howatt, *chairman,* J. F. Hale, H. M. Hart, E. V. Hill and S. R. Lewis to present this matter to the Council and secure approval.

S. I. Rottmayer was then introduced by President Aeberly and addressed the meeting on the subject, Forced Circulation Hot Water, discussing its adaption to various types of heating, cooling and dehumidifying, and to non-ferrous piping, building insulation, pipe insulation, and electric pneumatic control. Slides and data from the actual installations were shown. Mr. Rottmayer handled his subject with remarkable clarity, presenting a precise and thorough dissertation.

C. W. DeLand called attention to the Semi-Annual Meeting of the Society at Milwaukee, June 1932, and discussed the program, attendance, entertainment, etc.

Massachusetts

May 23, 1932. About 50 members and guests listened attentively to W. L. Fleisher at a luncheon meeting of the Chapter held at Durgin Park's, Boston. Mr. Fleisher read a paper entitled, Vacuum Steam Refrigeration and outlined the principles of steam jet refrigeration as applied to air conditioning for comfort. At the close of his reading, Mr. Fleisher competently answered many questions.

The election to the Board of Governors for a term of three

RESEARCH AND PROGRESS

G. L. LARSON, CHAIRMAN
Committee on Research

PROBABLY the most interesting research project being undertaken by the present A.S.H.V.E Committee on Research is the Summer Cooling investigation in the Research Residence at the University of Illinois.

This work, sponsored by the Society in cooperation with the *National Warm Air Heating Association*, is being carried on under the direction of the A.S.H.V.E Technical Advisory Committee on Refrigeration in Relation to Air Treatment, of which Prof. A. P. Kratz is chairman. Special funds amounting to $2500.00 have been contributed by two large organizations interested in this work and it is probable that contributions will be made by others as the work progresses. Because the work can be carried on effectively only during the summer, the scope of the investigation in any one season will be somewhat limited, but the following objectives have been planned for the Summer of 1932:

1. Tests with the entire residence cooled and the awnings in place and lowered. Cooling to be done by the central system now being installed in the residence.

2. Tests with unit coolers in individual rooms with the remainder of the residence cooled as in (1) and with awnings in place and lowered.

3. Repeat tests in (1) and (2) with awnings removed.

4. Tests with residence cooled with night air.

5. Repeat some of tests in (1) or (3), simulating cooking operations in the kitchen.

6. Tests on cooling the residence with an air washer using several water temperatures.

7. Tests with cool air from the basement circulated by means of the duct distributing system.

Such work as this is fundamental to the progress of summer air conditioning.

The Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is supported by a share in the dues of the members, plus a share in the profits of THE GUIDE, plus gifts from its friends.

The finances of the Laboratory are budgeted, expenditures are safeguarded carefully, and are audited each year by a certified public accountant.

Contributors who have placed their names on the Honor Roll so far in 1932 are listed below:

Aerofin Corp., Newark, N. J.
American Radiator Co., New York, N. Y.
Barber-Colman Co., Rockford, Ill.
Barnes & Jones, Boston, Mass.
Bush Manufacturing Co., Hartford, Conn.
Carrier Engineering Corp., Newark, N. J.
Celotex Co., Chicago, Ill.
Crane Co., Chicago, Ill.
Detroit Edison Co., Detroit, Mich.
Detroit Stoker Co., Monroe, Mich.
Frigidaire Corp., Dayton, Ohio
General Electric Co., Schenectady, N. Y.
Hoffman Specialty Co., Waterbury, Conn.
Howard Iron Works and
Alberger Heater Co., Buffalo, N. Y.
Johns-Manville Corp., New York, N. Y.
Johnson Service Co., Milwaukee, Wis.
Leeds & Northrup Co., Philadelphia, Pa.
Herman Nelson Corp., Moline, Ill.
Owens-Illinois Glass Co., Toledo, Ohio
Petroleum Heat & Power Co., Stamford, Conn.
Powers Regulator Co., Chicago, Ill.
Trane Co., La Crosse, Wis.
United Engineers and Constructors, Philadelphia, Pa.
Warren Webster & Co., Camden, N. J.
Youngstown Sheet and Tube Co., Youngstown, Ohio
W. H. Carrier, Newark, N. J.
American Oil Burner Association
Heating & Piping Contractors National Association
National Association of Ice Industries
Ventilating Contractors Employers Association

years of E. W. Berchtold and L. D. Cushman was announced by the secretary. The retiring members of the Board are R. T. Kern and Pres. David Moulton, who remains an ex-officio member.

Vice-President W. T. Jones of the National Society urged all who could to attend the Semi-Annual Meeting of the Society at Milwaukee, June 1932.

At the close of the regular Chapter meeting, the Board of Governors met for the annual election of officers. F. R. Ellis was elected Chapter president and Leslie Clough, secretary and treasurer for the ensuing year.

Those attending were Messrs. Ellis, Swaney, Berchtold and Clough of the Board of Governors; Past-President Moulton and Messrs. Jones, Boyden and Kellogg.

Philadelphia

May 12, 1932. This meeting was held at the Llanerch Country Club, Llanerch, Pa. On account of the inclement weather it was necessary to postpone the scheduled golf meet until Thursday of the following week.

Dinner was served at the club house, after which the meeting was called to order by Pres. L. C. Davidson. The minutes of the previous meeting were read and approved. The Treasurer's report was submitted and approved as read.

C. V. Haynes discussed the Semi-Annual Meeting of the Society to be held at Milwaukee, and urged a large attendance from the Philadelphia Chapter.

A rising vote of thanks was extended to J. H. Hucker, chairman of the Meetings Committee, for his splendid work in preparing a series of most interesting meetings. Harry Black spoke regarding the good work being carried on by the Salvation Army.

The meeting was then turned over to Mr. Hucker, chairman of the Entertainment Committee, and the balance of the evening was given over to the entertainment, at the conclusion of which the meeting was adjourned.

Western Michigan

May 16, 1932. The Spring frolic and annual meeting of the Western Michigan Chapter took place in the afternoon and evening of May 16, 1932, at the Masonic Country Club, Grand Rapids. The Program Committee consisting of Joe Troske, Don Taze, Karl Ziesse and Harry Christenson saw to it that frivolity was the order of the day.

Golf was the feature of the frolic, in which the Masonic hills and dales reverberated the shouts and cat calls of the gallery and the disappointed eliminated players. Joe Troske demonstrated his coolness by eliminating his first opponent on the ninth hole. (There ought to be a law against Program Committee Chairmen winning their own game). However, Joe well earned the honors bestowed on him.

Awards were also made to Harry Christenson as runner-up, to Pete Wierenga, best approacher, to Walter Pearl, low gross score and to O. D. Marshall, best dressed golfer. Bill Bradfield and Con Lammers won the horse shoe tournament uncontested.

The following were unanimously elected to office at the evening business session, and were inaugurated amid several jovial, but sincere addresses:

President—D. L. Taze
Vice-President—K. L. Ziesse
Treasurer—S. H. Downs
Secretary—P. O. Wierenga
Board of Governors—C. Lammers, O. D. Marshall, C. H. Morton.

President-elect Taze presented to retiring President Marshall, a framed embossed copy of a resolution of the Board of Governors in commendation of his administrative work during the first year of the Chapter's existence.

The year closed with the Chapter in a healthy financial co-

dition, an enthusiastic and loyal membership body, and every assurance of stability and growth. Under the leadership of President Taze, a well balanced and successful year is assured.

Michigan

May 10, 1932. After spending the afternoon at golf and baseball, about 65 members and guests gathered for dinner in the dining room of the Meadowbrook Golf and Country Club. After the dinner, Pres. G. H. Giguere asked for the roll call and the reading of the minutes of the last meeting.

Tom Brown announced the names of those who performed unusual feats at the golf meet as follows:

E. J. Anderson—1st Low Gross Score
H. W. Sherrif—2nd Low Gross Score
J. Trosky—3rd Low Gross Score
Mr. Hull—Longest drive from 1st tee
G. D. Winans—Shortest drive from 1st tee
A. C. Wallich—Highest Gross Score

also Messrs. Dellis, Arnoldy, Fisher, Deppman, Trinkhouse, Brown, McConachie, Robinson, Nelson, Sterner, Smith, Sheely, McFait, Billel, Nannen, Walton, Glanz and Rowe.

W. G. Boales reported on the Semi-Annual Meeting 1932 to be held at Milwaukee. E. H. Clark reported on the amount of money in the treasury and President Giguere appointed an auditing committee consisting of E. E. Dubry, L. H. Kintz and Mr. Harrigan.

The meeting was addressed by W. Long of Euclid Investors, Inc., on America's Future, after which Jack Rush entertained the meeting by telling several Irish stories.

The officers to be elected were submitted to the members by the Nominating Committee, and nominations were called for from the floor. No nominations were made, and it was moved and seconded that the nominations be closed. J. R. McColl, J. L. Fuller and Edward Glanz acted as tellers. The following were elected:

President—L. L. McConachie
Vice-President—H. E. Paetz
Treasurer—E. H. Clark
Secretary—G. D. Winans
Board of Governors—A. C. Wallich, J. W. Snyder, R. K. Milward.

Mr. Giguere then turned the meeting over to Mr. McConachie.

St. Louis

May 4, 1932. This dinner meeting of the St. Louis Chapter was held at the Roosevelt Hotel and was attended by 22 members and guests.

Pres. J. M. Foster called the meeting to order and the minutes of the March and April meetings were read and approved. A letter was read from C. A. Pickett, chairman of the Entertainment Committee, suggesting that the June meeting be set aside as the annual outing. A motion was made by E. A. White and seconded by K. S. Wallace, that the June meeting be held at the Meadowbrook Country Club, June 4th, and that a charge of $2.00 per plate be made to cover the actual cost of the dinner. Motion carried.

In the report of the Limited Chapter Membership Committee, the application of G. S. McClosky was presented.

L. Walter Moon, Secretary of the Code Committee, reported that a code on heating and ventilating had been prepared by his committee and turned over to the *Associated Engineering Society*. A copy of this code was presented to the Secretary to become a part of the minutes of this meeting.

President Foster then introduced Professor Fitch of the Architectural School of Washington University. Mr. Fitch spoke to the Chapter on the course he is conducting at Washington University on heating and ventilating and requested that the members of the Chapter co-operate with him in presenting extra copies of THE GUIDE which is used as a text book.

The meeting was then turned over to Paul Sodemann, chairman of the Program Committee, who introduced Prof. A. S. Langsdorf, Dean of the School of Engineering, Washington University. Professor Langsdorf talked on dust counting and described the various methods used in securing data on this subject, as well as soot fall. He then submitted information on the use of a photo-electric cell for dust counting and for determining soot fall. With this method, dust counting and soot fall can be accurately determined graphically within the radius of a mile. Through a very ingenious hook-up of the electric projector, it is possible to eliminate any secondary light. Professor Langsdorf then made a practical demonstration of the photo-electric cell in the determination of dust counting.

Through the courtesy of the Union Electric Light and Power Company a talking picture was presented on the subject of Lake of the Ozarks. This picture describes the Bagnell Dam project.

A vote of thanks was given to Professor Langsdorf and also to H. W. Drissell of the Union Electric Light and Power Company, for the splendid paper and the excellent entertainment.

Southern California

May 10, 1932. The last meeting of the season of the Southern California Chapter was held at the Engineers' Club, Los Angeles. After dinner was served, President Winch called the meeting to order and reports from the various committees were submitted. He then appointed the Nominating Committee for 1933 consisting of C. M. Barker, *chairman*, Carl Johnson, O. W. Ott, Carl Thomas, Andrew Orear.

The meeting was turned over to L. H. Polderman, chairman of the Program Committee, who introduced S. B. Carpender, vice-president of the Carrier Corp., and first vice-president of the Brunswick-Kroeschell Co., who related some interesting experiences.

Mr. Polderman then introduced Major G. L. Cheney, Reserve Officer of the Chemical Warfare Division of the United States Army, who presented, under the auspices of that group, a talk and motion pictures on the Chemical Warfare and the effect of gases on the atmosphere. This was very interesting, and quite different from the usual technical subjects discussed at our meetings. It is recommended that the various Chapters get in touch with the Chemical Warfare Division and endeavor to arrange similar programs.

Western New York

May 9, 1932. The May meeting was held at the New York Telephone Building with about 35 members and guests present. After dinner the meeting was called to order and the minutes of the last meeting accepted.

Pres. M. C. Beman then took up the question of get-togethers with neighboring Chapters and after some discussion he recommended that the Program Committee attempt to arrange such a meeting for the fall.

It was also decided that the Chapter could do nothing towards financially helping the Engineering Unemployment Fund but that private subscriptions would be most welcome.

Joe Davis then introduced the question of the Annual June Dinner Dance and it was decided to hold this as usual at the Auto Club but at a much lower cost.

The Semi-Annual Meeting of the Society at Milwaukee was then discussed and several members signified their intention of attending.

A. C. Coupe, Public Relations Department of the New York Telephone Co., was then introduced. An interesting moving picture showing how transatlantic calls are made was shown. The members were then conducted in groups through the maze of mechanisms making up their new automatic dial system. This inspection trip was one of the most interesting and instructive sessions of the season.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 15 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some members by July 15, 1932, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
BARBERA, HENRY A., Student, New York University, New York, N. Y.	R. D. Morrill C. P. Bliss—*ASME*	C. E. Gus—*ASME* W. R. Bryans—*ASME*
BRAUN, JOHN J., Factory Mgr., U. S. Playing Card Co., Cincinnati, O.	K. A. Wright R. R. Taliaferro	F. D. Mensing E. N. Sanbern
FAHNESTOCK, M. K., Special Research Asst. Prof., Univ. of Ill., Urbana, Ill. (*Reinstatement*)		
FREITAG, FRED G., Consultant, Sylvestre Oil Co., Mt. Vernon, N. Y.	O. O. Oaks T. W. Reynolds	A. L. Jones C. A. Mehne
HAJEK, WILLIAM J., Br. Mgr., Minneapolis-Honeywell Regulator Co., Boston, Mass.	John Turner A. F. Karlson	Leslie Clough D. S. Boyden
HESS, DAVID K., Student, University of Wisconsin, Madison, Wis.	J. E. Mack—*Non-Member* W. F. Steve—*Non-Member*	M. H. Hebb—*Non-Member* G. L. Larson
HOPPER, GARNET HENRY, Engr., Taylor-Forbes Co., Ltd., Toronto, Can. (*Reinstatement*)		
JONES, DAVID J., Control Engr., Mgr. Accessory Div., Holland Furnace Co., Holland, Mich.	E. V. Hill P. J. Marschall	V. W. Cherven
MAIMAN, HERBERT, Student, New York University, New York, N. Y.	R. D. Morrill C. E. Gus—*ASME*	W. R. Bryans—*ASME* C. P. Bliss—*ASME*
MOTZ, O. WAYNE, Mech. Engr., Samuel Hannaford & Sons, Cincinnati, O.	C. J. Kiefer G. B. Houliston	W. C. Green
O'HARE, GEORGE W., JR., Student, New York University, New York, N. Y.	R. D. Morrill W. R. Bryans—*ASME*	C. P. Bliss—*ASME*
PECK, ROBT. E., Vice-Pres., Peck, Hannaford & Peck Co., Cincinnati, O.	C. J. Kiefer G. B. Houliston	W. C. Green
TENKONOHY, R. J., Vice-Pres., Airtherm Mfg. Co., St. Louis, Mo. (*Reinstatement*)		
TREADWAY, J. QUENTIN, Sales Engr., Clarage Fan Co., Toledo O.	C. R. McConner S. H. Downs	H. C. Baker
WINTHER, ANKER, Air Cond. Engr., York Ice Machinery Corp., Cincinnati, O.	J. J. LaSalvia L. J. DuBois	J. R. Hertzler M. Kennedy

Candidates Elected

In past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Membership Committee and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

ADLAM, T. NAPIER, Chief Engr., Sarco Co., New York, N. Y.

BRAY, HARRY FRANKLIN, Sales Engr., Excelsior Stove & Mfg. Co., Quincy, Ill.

COOPER, JOHN WILLIAM, Dist. Repr., Buffalo Forge Co., St. Louis, Mo. (*Advancement*)

GOEZ, WALTER E., Pres., Old Orchard Sheet Metal Works, Webster Groves, Mo.

MAEHLING, LEON S., Equitable Gas Co., Pittsburgh, Pa.

SCHNEIDER, WILLIAM G., Copper and Brass Research Association, New York, N. Y.

SHEPARD, EDWARD C., Shepard Engineering Co., New York, N. Y.

SUTTON, FRANK, Consulting Engr., 140 Cedar St., New York, N. Y.

VROOME, ALBERT EDGAR, Service Engr., United Engineers & Constructors, Inc., Philadelphia, Pa.

ASSOCIATES

MOODIE, DWIGHT L., Sales Engr., A. K. Howell Co., Inc., St. Louis, Mo.

MOREAU, DONATO, Mgr.,, Htg. & Vtg., J. Knox Corbett Lumber Co., Tucson, Ariz.

REED, PAUL L., Editor, Warm Air Heating, St. Louis, Mo.

WILSON, WILLIAM HENRY, Pullman Car & Mfg. Corp., Chicago, Ill.

JUNIORS

BARKER, HUGH PURSLOVE, Pres., Barker, Young & Co., Ltd., Westminster, England

LANDAUER, LEO L., C. L. Kribs, Jr., Cons. Engr., Dallas, Texas

STUDENT

LOWE, HOWARD HENRY, Student, Univ. of Minnesota, Minneapolis, Minn.

Heating · Piping and Air Conditioning
EDITORIAL

PREPARING *FOR* TOMORROW

An engineer from South America tells of parts of that continent where only the very wealthy have a miniature stove for heat, and two candles for light. The lives of most of the people are guided by emotion, superstition and mysticism. They plant infertile seeds and blame the moon. Those with ambition aspire to become poets, writers, artists, orators, soldiers, or politicians,—ordinary work is not honorable.

Those who read wonder at the advancement of North America. They marvel at the way in which we harness the physical sciences in a power house and do more work than all their beasts of burden. They do not understand that our material advancement is the result of the engineering type of mind,—the discovery of facts by testing.

But as a nation our engineering method of thought so far has been applied to material things. The present paradoxical economic situation would indicate that in business we have been planting infertile seeds and blaming the moon. It is now proposed that industry help to prepare for tomorrow by applying its own engineering method of thought to social and economic problems. When this is done it is possible that we may look back upon the present illogical situation as we look upon those parts of the world where the wealthy use a miniature stove and burn two candles.

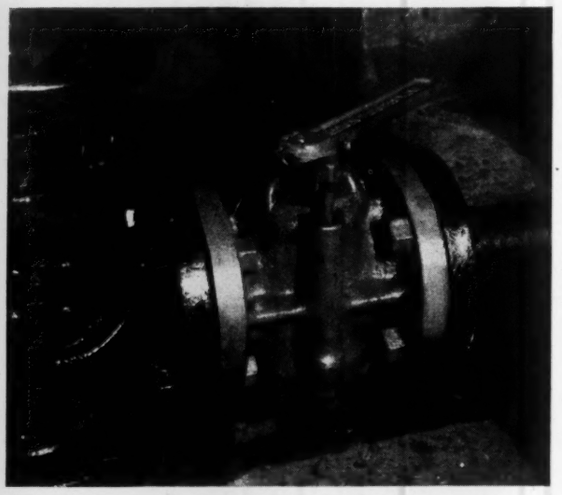
INTERESTING Equipment . . . Developments

A Plug-Type Valve

DESIGN of a plug type lubricated valve has been improved by the American Car and Foundry Company, 30 Church St., New York City, and it is now manufactured with a solid head construction to avoid the use of glands, gland bolts, bonnets or stuffing boxes; no packing or gaskets are employed. The valve opens and closes with a quarter turn against a positive stop; there are no seats. The face-to-face dimensions of the flanged valves are the same as standard gate valves.

A semi-steel, nickel-chromium composition (a close-grained, dense metal of high tensile strength) is used for the making of castings for the extra heavy duty valves.

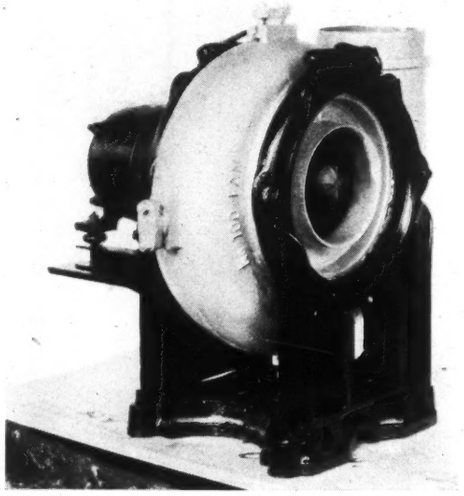
An example of the type of service on which these valves are now used is the handling of hot rosin at 580 F,



where a pressure of 30 lb. is maintained on one side of the valve with a 30-in. vacuum on the other. The illustration shows a 2½-in. valve installed on a boiler blow-off line. It is between a gate valve and the blow-off line, as most state laws require two valves in this service.

Fans for Fumes

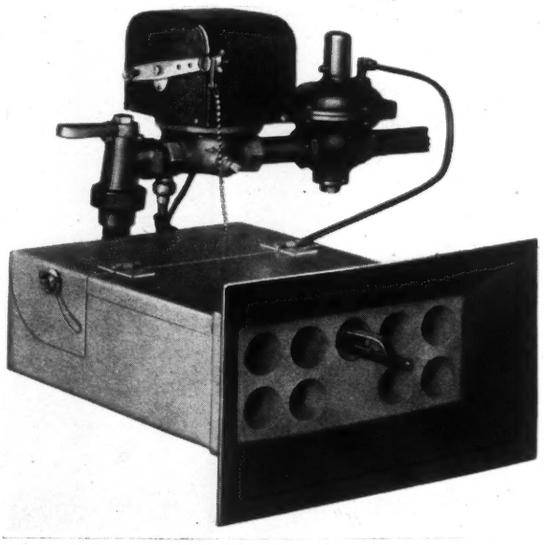
A NEW line of exhaust fans for handling corrosive fumes has been recently developed; it will be noted from the illustration of the 8-in. fan shown here that the housing is of one piece; previous models had split housings. All moving parts are cushioned on rubber, the impeller being hung directly on the motor shaft and the motor mounted on rubber cushions. In fan sizes 8-in. and up, ball-bearing motors are used; smaller sizes are equipped with sleeve bearings. All parts which touch fumes are made of a special alloy which does not require coating or painting. Design is such that all moving parts may be withdrawn from the fan without disturbing the duct connections. These fans are made by The Duriron Company, Inc., of Dayton, Ohio.



New Conversion Units

THE gas conversion unit illustrated consists of a burner encased in a steel jacket which is provided with an air damper for manual or automatic operation, an automatic electric motor valve, safety main and pilot light valves, and a pilot light with a safety switch.

These conversion units use the multi-unit, multi-jet principle; multiple mixing chamber and multiple gas jets for each mixing chamber, in order to produce



thorough mixing of gas and air to promote complete combustion to the end of the burner with minimum excess air and high furnace temperatures. Combustion occurs in a refractory mixing block over the end of the burner near the manifold. The units are assembled complete with pipe and fittings; two assemblies include a gas pressure regulator with copper vent pipe, room thermostat and armored cable.

An improved safety switch is incorporated with the unit and consists of a pilot light tube with thermostatic element, a base containing gas supply pipe and nozzle for the pilot light, electrical terminals and stationary contact point, and an outer safety cover. The heat from the pilot flame is the actuating force on the thermostatic element; the pilot light serves to ignite the main gas burners and to operate the automatic safety switch. The manufacturer is Lee B. Mettler Co., 406 S. Main St., Los Angeles, Calif.

Small Steam Traps

AN ADDITION to its line of steam traps has been announced by the V. D. Anderson Company, Cleveland, Ohio. The trap is light in weight and small in size; it requires little space for pipe connections thereby making it particularly applicable for use on sterilizers, pasteurizers, blanket warmers, coffee urns, dish heaters, jacketed kettles, laundry presses, heating coils, unit heaters and many other types of small steam-using units. The trap can be furnished chrome plated and highly polished.

The working parts are made of heat-treated stainless steel. The removal of one bolt enables the trap to be

taken apart for inspection without disturbing the pipe connections. The trap may be used either as an elbow or straight in line. It is regularly furnished with ½-in. pipe connections, however, ¾-in. connections are furnished when desired.

Capacities are as follows:

Operating pressure—lb. .	5	10	15	20	30	40	50
Discharge lb. per hr.	475	600	700	750	450	500	600
Operating pressure—lb.		60	70	80	90	100	125
Discharge lb. per hr.		500	550	600	500	525	575

Net weight, 3¼ lb.; dimensions—diameter 2⅞ in., height approximately 5 in.

Room Cooler of Interest

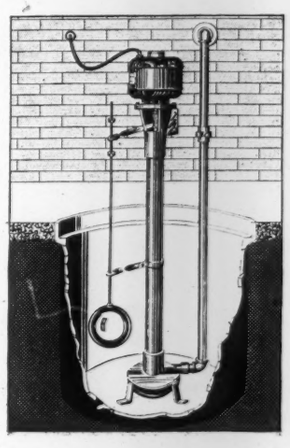
THE portable room cooler illustrated here uses ice as a cooling medium; by means of a fan warm air from the room is drawn into the cabinet through an intake grille and passes over the cooling coils, through which ice water is pumped. A control governs the amount of water which passes through the coils, which is circulated by a bronze water pump

from the bottom of the cabinet through the coil and back to the bottom of the cabinet.

The unit, which has a capacity of 300 lb. of ice, is 50 in. high, 30 in. wide and 22 in. deep. A 1/6-hp. motor drives the fan and the pump. The cooler is iced through a service door, and is provided with rubber-tired wheels for portability. According to the manufacturer (Chicago Pump Company, 2336 Wolfram St., Chicago) one of these units maintained a room 20 ft. by 21 ft. by 11 ft. containing five people at 81 F with an outside temperature of 95 F.

For Pumping the Sump

CAPACITY of a new automatic electric sump pump ranges from 3,000 gallons per hour at low heads to 1,000 gallons at the maximum head of 20 feet.

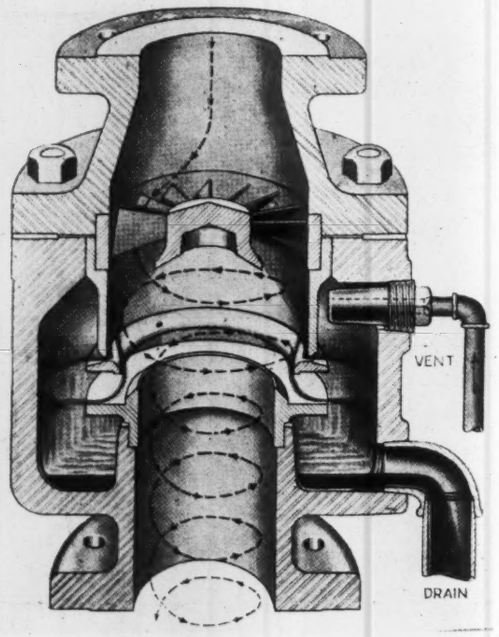


The motor is ¼ hp. heavy duty, ball bearing, vertical type, designed especially for sump pump application. The pump is driven through a stainless steel shaft supported by oilless bearings; a ball

bearing thrust is provided. The pump is of all bronze construction having a semi-enclosed impeller designed to handle ordinary solids. It is made by the M. L. Oberdorfer Brass Company, 8002 Thompson Road, Syracuse, N. Y.

Removes Oil from Steam

A LINE purifier of the centrifugal type is especially recommended by the manufacturer for removing oil from exhaust steam; in many plants exhaust from reciprocating units is wasted because of oil or foaming



trouble may be experienced with water in the boiler. Removal of oil from steam is important when the steam is to be used for process work and desirable when it is to be used for heating.

The device may be installed in horizontal pipe lines or in vertical lines for up or down flow; it has no moving parts, the centrifugal motion being imparted to the steam, gas or air by means of a number of stationary tuyere blades. In the illustration, the dotted line represents the path of a steam jet; the drain leads to a trap which is vented back to the purifier as shown.

Housing for the 125-lb. purifier is of cast iron; cast steel is used for the 250-lb. and 400-lb. constructions. The manufacturer is the Centrifix Corporation, Cleveland, Ohio.

Solder for Aluminum

AN aluminum and all-metal solder, capable of withstanding up to 500 lb. steam pressure at live-steam temperature, has recently been announced. It has a tensile strength of 12,000 lb. per sq. in. The solder is applied with an ordinary soldering iron or blow torch, melting and fusing at a temperature of 370 F. Once having melted, releasing the catalyzer from the solder, it will withstand a considerably higher temperature. The solder is hard enough to machine easily, is ductile, and chromium, nickel, or any other kind of plating may be applied over it.

Test samples are available from the Allied Research Laboratories of Glendale, Calif.

For Cooling Rooms

A PORTABLE air-cooling unit using melting ice as the cooling medium has recently been perfected and is designed for use in offices and committee rooms, hotels, hospitals, small stores, beauty parlors and barber shops, tea rooms, etc. The device will lower the temperature of a room approximately 10 degrees from, say, 80 F to less than 70 F. As the hot room air is cooled by contact with the melting ice and a series of metal grids upon which the ice rests it is dehumidified.

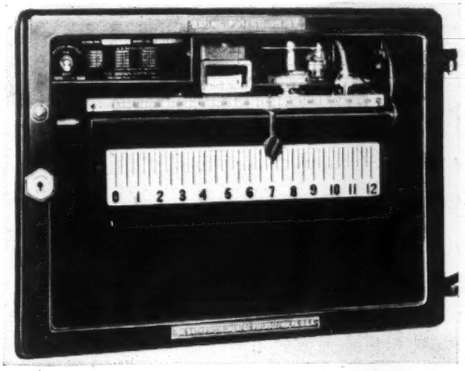
The unit consists of a cabinet on wheels standing 4 ft. 3 in. high, 2 ft. wide and made of sheet steel finished in grained mahogany. The weight of the cabinet, without ice, is 400 lb.; it holds 300 lb. of ice, sufficient to last about five hours in hot weather. A small electric-driven blower operated from a light socket draws the air from the room into the cabinet and over the metal surface on which the ice rests, thence back into the room through outlets in the top of the cabinet. Amount of cooling and direction of air delivery can be regulated by adjusting a set of shutters in the outlets. Capacity is 400 c.f.m. of cooled air.

This development is announced by the Carrier Research Laboratories, Newark, N. J.

Potentiometer is Self-Balancing

A NEW indicating potentiometer is of particular interest because of its self-balancing feature, The instrument has a 40-in. slide wire, positive spiral drive, a secondary pointer which fixes the size of step in moving the slide-wire contact and mercury-in-glass switches in control models. The mechanism is arranged to swing out of the case for inspection without interference with operation.

The large scale and figures, calibration marks and pointers are designed to facilitate accurate reading at

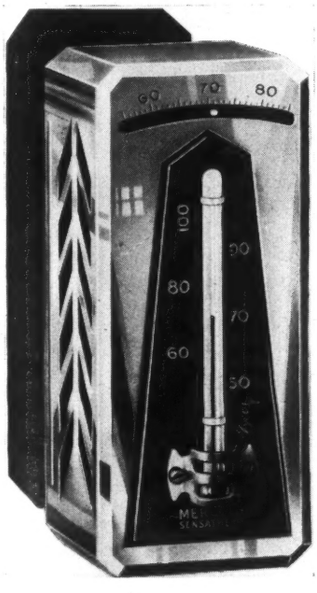


considerable distances. As a pyrometer the instrument is supplied for any range from 0 to 100 F up to 0 to 3,000 F or for most spans between these ranges; models are furnished for automatically controlling as well as indicating temperatures; the manufacturer is The Brown Instrument Company of Philadelphia, Pa.

Controls Air Temperature

THIS thermostat is a low-voltage air-temperature control designed to operate in connection with a transformer relay; on alternating current the voltage is under 24 volts and when used in connection with a repulsion relay, the same instrument can be used on d-c. It was designed with particular attention paid to sensitivity, compactness and neatness.

It is operated by means of a spiral bimetallic element sensitive to changes in temperature. One end of the element is attached to the range adjustment lever and the other end is free and moves a small permanent magnet attached to it. When the temperature of the bimetallic element goes below the temperature for which the device is set, the magnet is held in close proximity to the switch, which contains a fixed set of contacts and a movable armature pivoted at its upper end and having a cup containing mercury at its lower end, all hermetically sealed in glass. The movable armature in the switch is drawn toward the magnet and the mercury cup is in such position so that the mercury closes the circuit between the fixed contacts. As the temperature increases, the bi-metallic coil moves the magnet away from the switch.

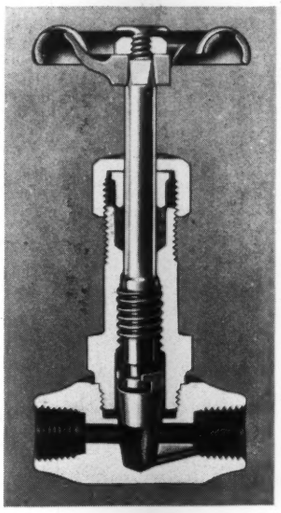


When the pull of the magnet can no longer hold the armature, the mercury cup moves away from the contacts and the circuit is opened. At the same time the armature swings back, the pull on the magnet is released and it goes back into position where it no longer affects the armature. Contact is closed and opened with a positive snap action.

Standard range is from 55 to 85 F and it operates on a temperature variation of ½ deg. above or below the point set; this total differential of 1 deg. can be increased to 2 deg. by an adjustment. The instrument is made by The Mercoid Corporation, 4201 Belmont Ave., Chicago.

This Valve Is Cleanable

A NEW valve design illustrated here is particularly recommended for refinery and other services where coking or similar troubles are encountered, as it can be cleaned by running a rod through the line. The valve is precision turned from a high-quality carbon steel and the stems are specially treated for high-temperature service. The renewable slip-on disc for stainless steel is free on the stem to prevent sticking in the body and galling of seating surfaces; it is made non-rotatable to prevent horizontal scratching of the seating surfaces. When wide open the disc is entirely out of the fluid passageway. A relief passage for the escape of fluids caught under the plug as it closes is provided. The body, bonnet, packing nut, and gland are treated for protection against corrosion.



The manufacturer is Reading-Pratt & Cady Company, Bridgeport, Conn.

Capacitor-Type Motors

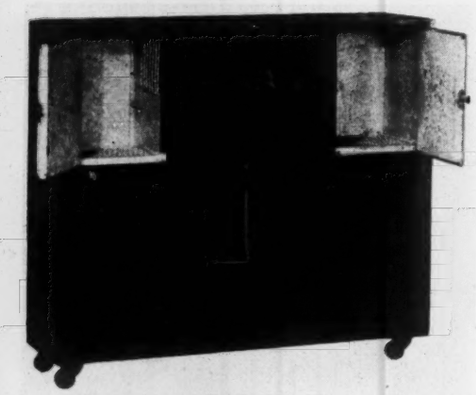
APPLICABLE to oil burners, refrigerators and similar devices are the capacitor motors recently made available, and one of which is illustrated here. They are condenser - start condenser - run motors, condenser capacity being varied from start to run by means of a switch and an auto - transformer. A box mounted as an integral part on top of the motor contains the condenser and transformer.

These motors are available rigid-mounted or rubber-mounted in drip proof or totally enclosed types. They are made by the Wagner Electric Corporation, 6400 Plymouth Ave., St. Louis, Mo.

Cools and Dehumidifies

THE portable air cooler shown here has a capacity of 300 lb. of ice in flake or block form and delivers 500 c.f.m. of cooled air; two multiblade-type fan wheels each handle 250 c.f.m. The picture shows the doors open for re-icing.

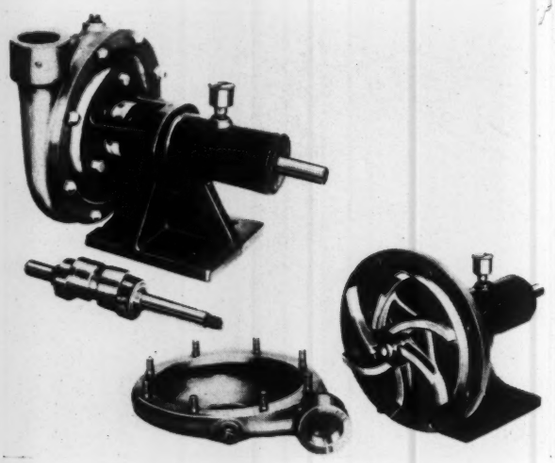
Air is drawn into the device at the bottom and



discharged through adjustable louvres through the top. Two types are available, one cooling the air by means of special radiators through which cold water is circulated and the other cooling it by sprays of cold water. In winter the equipment may be used to heat and humidify air. The cabinet, which is insulated, is mounted for portability on rubber-tired casters; it is 50 in. long, 46 in. high and 15 in. wide. Koolair-Maker Corporation, 4485 Olive St., St. Louis, Mo., is the maker.

Handles Chemicals, Foodstuffs

BALL-BEARING centrifugal pumps recently developed for handling chemicals, liquid foodstuffs and other uses are designed to provide large capacity for each horse power used with minimum maintenance cost.



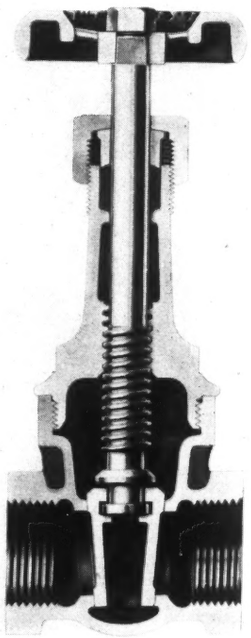
Shaft and impeller alignment is accomplished by providing ball bearings to carry the load in any direction. The stuffing box is of the split screw gland type; there is no bearing in back of the packing, which is said to eliminate possibility of lubricant entering the pump and mixing with the liquid handled.

The pumps have removable inboard and outboard heads; when it is necessary to have equipment that can be quickly taken apart for cleaning, the nuts that attach the inboard and outboard heads are replaced with large wing nuts that can be tightened without the aid of tools.

The manufacturer is Hills-McCanna Company, 2349-59 Nelson St., Chicago, Illinois.

Valve for 125-Lb. Steam

A NEW line of rising stem bronze gate valves for 125-lb. steam pressure and 175-lb. water pressure have been made available. A flexible connection is provided between the disc and the stem so that the stem will not bind or spring when the valve is closed. The stem is made of a bronze composition of high tensile and torsional strength, and has a large number of contact threads in the bonnet.



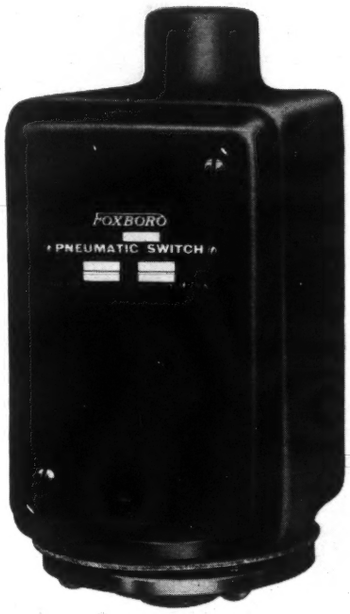
The bonnet hexagon is extra large and close to the body to prevent distorting the bonnet when a wrench is applied for installing or disconnecting. Other features are the deep stuffing box with gland and square section molded packing rings, the large malleable iron handwheel secured to the stem by a nut, and the heavy, wide pipe-end hexagons with large number of threads.

These valves are announced by the Kennedy Valve Mfg. Co., Elmira, N. Y.

Pneumatic Switch Is Compact

A COMPACT pneumatic switch has been designed by The Foxboro Company, Foxboro, Mass., for use with its standard temperature and pressure controllers. This switch makes possible the use of air-operated controllers on equipment that is controlled by turning on and off an electric current. It has been used successfully to control temperatures in electrically-heated ovens, to control motor-operated pumps, blowers and compressors, and as a means of producing audible or visible signals on time cycle applications.

The switch is a standard single-pole mercury type

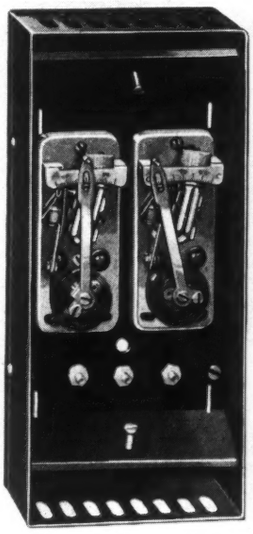


switch mounted in a small cast-iron housing with an outlet tapped for either conduit or BX connector. In conjunction with suitable auxiliary magnetic switches, it may be used in any circuit where the current is not higher than 600 amperes at 220 volts. The switch is operated by a small diaphragm motor mounted directly on the housing. The standard switch is closed by air pressure from the control instrument and opened by a spring. The switch is always open, therefore, when the air is off.

An Infiltration Thermostat

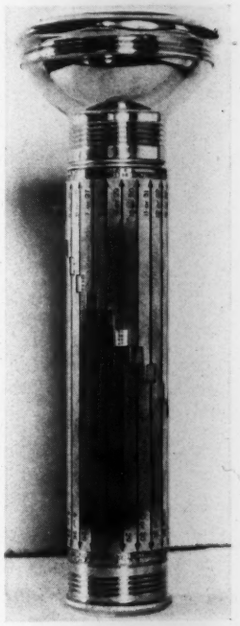
A NEW type of thermostat designed to control the temperature in a room on the basis of temperature plus infiltration has been developed and a rear view of it is shown here. The instrument is mounted at radiator height on an outside wall. The dampers which appear in the cut are adjustable and are intended to control the amount of air rising up through the thermostat; adjustment is made when the control is installed and varies with the characteristics of the room and the wall construction.

There are two types—electrical and pneumatic, according to J. C. Hornung, 343 S. Dearborn St., Chicago, who is the manufacturer of these infiltration thermostats.



For the Dry-Kiln Operator

A DEVICE designed to be of aid to the dry kiln operator or supervisor of a drying oven is the combination flash-light and relative humidity determinator shown here. The light is useful in reading instruments or for inspecting stock while in the process of drying.



When wet- and dry-bulb readings are noted a twist of the revolving outer sleeves will show the relative humidity for a range in temperature from 100 to 200 F and with any wet-bulb depression reading up to 40 degrees on any point within this range. Figures and arrows are in two colors to aid reading and are etched in brass. J. B. Maas & Company, Fox Creek Station, Detroit, Mich., is the manufacturer.

The light is of a type used widely for outdoor work and operated by three dry cells.

Recent Trade Literature

Air Conditioning: York Ice Machinery Corporation, York, Pa.; 16-page folder describing the advantages of air conditioning restaurants, and showing several air-conditioned restaurants and the equipment; a 20-page bulletin on refrigeration and air conditioning for railroad passenger cars, describing individual units driven electrically and units driven by gasoline engine; a four-page folder describing a milk room cooler; a four-page bulletin describing a unit cooler for fur storage, candy manufacture, florists, offices, etc., including a diagram of a complete system.

Air Filtering (Industrial): American Air Filter Company, Inc., Louisville, Ky.; 28-page booklet on industrial dust problems and the use of filtered air for atmospheric dust control, industrial air conditioning, industrial ventilation, drying operations, product finishing, control of bacteria and mold spores, cooling electrical equipment, protection of engines and compressors, control of process dust, and miscellaneous industrial applications. Typical examples are given.

Alloy Steel: The International Nickel Company, Inc., 67 Wall st., New York City; 12-page article giving data and applications for nickel alloy steel castings.

Blowers and Compressors: Roots-Connersville-Wilbraham, Connersville, Ind.; four-page pamphlet describing the features of single- and multi-stage centrifugal blowers and compressors for handling air or gas under pressure or vacuum. Characteristics and capacities are given.

Boilers: Kewanee Boiler Corporation, Kewanee, Ill.; four-page bulletin containing engineering information and a description of a welded round steel heating boiler